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H. W. ALDEN, *President* COKER F. CLARKSON, *Secretary* C. B. WHITTELSEY, *Treasurer*

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THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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Chronicle and Comment

July 1923 Issue of Data Sheets

THE July issue of data sheets will be mailed to the members early in August. The issue will contain 138 pages, 50 of which are new, and is the second largest issue since the S. A. E. HANDBOOK was revised in 1920.

Instances have been brought to the attention of the Society where disagreement has been caused through the use of obsolete data sheets. Members would do well to have their books, after the new issue has been inserted, checked to the list of uptodate sheets and dates of issue that is included in the letter of transmission.

As the revised S. A. E. HANDBOOK will contain 558 pages, members may find it more convenient to use two binders, one binder being used for Sections A, B and C and the other for Sections D to K.

1906 Transactions Wanted

A COPY of the first volume of the Transactions of the Society, that for the year 1906, is desired to complete a set maintained as part of the archives of the Society. It will be appreciated if any member who has a copy of this volume and does not desire to keep it for his own library, will communicate with the offices of the Society in New York City.

Part II 1921 Transactions

THE second part of the TRANSACTIONS of the Society for the year 1921 was mailed recently to those members who placed orders for them. This volume, which completes the sixteenth issued by the Society, contained 20 papers given at Society meetings during the latter half of 1921 and included a total of 566 pages. The papers include the able treatise on gas-engine cylinder actions by Sir Dugald Clerk that marked the beginning of a greater appreciation of the value of combustion research in this country. A majority of the contents deals with various elements of the fuel problem, turbulence and combustion.

When this issue of THE JOURNAL went to press, the work of publishing the next volume of TRANSACTIONS, Part I, 1922, was well under way. Only enough copies will be printed to meet the requests of the members and no orders will be accepted for this volume after Sept. 4, 1923, when it will go to press. An order blank for the next volume of TRANSACTIONS, Part I, 1922, is included in this issue of THE JOURNAL on p. 54 of the advertising section.

Papers on Production Problems

THE Meetings Committee of the Society is particularly anxious to make the 1923 Production Meeting, which will be held at Cleveland, Oct. 25-27, at least as successful as the meeting at Detroit last October. The success of the Detroit meeting can be attributed largely to the excellence of the papers presented and the timeliness of the subjects chosen for discussion. To achieve success at Cleveland, it is recognized that current problems whose solution is urgent must again be chosen for treatment in the sessions. A canvass of production authorities is now being made to ascertain which problems are of greatest general interest. Suggestions are in order and will be received gratefully if forwarded to the office of the Society in New York City.

The interest in the problem of cutting accurate and quiet-running gears alone assured a large and appreciative audience once that topic was selected for study at the Detroit meeting. Gear-cutting still possesses the same attractiveness as a subject as it did last year and one session will be set aside for it at Cleveland. Manufacturers of gear-cutting machines are being asked to present papers recording what progress has been made in correcting the errors revealed by the 1922 meeting. Papers are being solicited from men who have been studying the causes of gear noise.

It is certain that more papers will be submitted than can be used at the Cleveland Meeting. Authors who are desirous of presenting papers can avoid disappointment by making their requests early and submitting outlines of papers on subjects that are known to be of general interest. Papers that attempt to do little but set forth the particular advantages of an individual machine are less likely to be accepted than those treating specific problems and presenting the experience and quantitative results of a carefully conducted piece of research in a reasonably unbiased manner. The probability of a paper's being accepted can be judged largely on the amount of practical workable information it presents for the attending production men to take back into the shop with them. The Society wants factory men to attend its meetings because they expect to carry home sufficient

helpful information to repay them for the time and expense of attendance.

Those present at the Detroit meeting can readily appreciate the prestige accruing to an individual and his employer from representation on the program of an important national meeting such as the Production Meeting. It is hoped that the privilege of representation on the Cleveland Meeting program will prove sufficiently attractive to influence many production authorities to submit papers in response to this invitation.

Men and Positions

THE list of Men and Positions Available that has always been published in the advertising section of THE JOURNAL has been discontinued with this issue. This step has been necessitated by improvements made in the employment service of the Society during the past year. Under the improved system, bulletins of positions available are sent semi-weekly to members seeking employment; similar bulletins of men available are mailed to a list of representative manufacturers and to companies making specific inquiries for men. These semi-weekly bulletins have filled the positions available with such dispatch that the monthly lists published in THE JOURNAL are often out-of-date before they reach the members. This condition has resulted in disappointments to several members who have written promptly to the office in New York City upon receipt of THE JOURNAL only to find that the position interesting them has been filled through the more frequent service of the bulletins.

Use of the bulletin service for a year has demonstrated clearly its superiority over the monthly lists in THE JOURNAL. Since THE JOURNAL lists can not be guaranteed as being uptodate while the semi-weekly bulletins are operating, it seems best to discontinue the publication of information that might be misleading.

Sectional Committee Reports on Screw Threads

THE Report of the Sectional Committee on the Standardization and Unification of Screw Threads has been submitted to the American Society of Mechanical Engineers and the Society of Automotive Engineers, the sponsor societies, and has been approved by the Council of the American Society of Mechanical Engineers and is being prepared by that Society in pamphlet form. The Sectional Committee was organized by the sponsors under the rules of procedure of the American Engineering Standards Committee, to review the Progress Report of the National Screw Thread Commission, issued as the Bureau of Standards Miscellaneous Publication No. 42, and to report thereon to the sponsors.

The report, which is intended for bolts, machine screws, nuts, commercial tapped holes and the like, includes sections treating of Terminology; Form of Threads; Series of Threads, including tables; Classification of Fits and Tolerances.

The Council of the Society at its meeting on June 18 at Spring Lake, N. J., assigned the report to the Screw Threads Division of the Society's Standards Committee with instructions to review it and report to the Standards Committee in accordance with the Standards Committee Regulations governing the acceptance and approval of Sectional Committee reports submitted to the Society as a sponsor. It is probable that the Division will accordingly act on the report as a whole for approval by the Society as Tentative American Standards, issued under the rules of the American Engineering Standards Committee and also report such parts of the Sectional Committee's report for adoption as S. A. E. Standards as are particularly applicable to general automotive practice. The report will be acted on in the same manner as the regular standardization work of the Society, and submitted to the Standards Committee and Society for approval at the meeting next January.

More on "Licensing" of Engineers

A GENERAL letter sent to the members recently called their attention to the fact that laws exist in a number of States providing in general that engineers doing public or private engineering work are required to register, paying fees in varying amounts up to $25 for such registration. Since sending this letter to the members, the office of the Society has been in communication with the secretaries of the various States endeavoring to secure copies of these laws and opinions on whether they affected automotive engineers. As a result, some corrections must be made in the list of States contained in the general letter to the members since, in contradiction of the original information received by the Society, some States do not require automotive engineers to apply for license.

The Secretary of State of West Virginia informs us that there is no law in that State covering the registration of automotive engineers. The Secretary of State of Illinois writes that "while bills were introduced in the present General Assembly relative to the licensing of mechanical and automotive engineers, these have not been passed and it is very doubtful if they will be passed during this session of the General Assembly." There is no State license required of professional engineers in Wisconsin, according to the Secretary of that State, attempts to pass such a law having failed at two sessions of the legislature. The State Comptroller of New Mexico informs us that there is no statute in that State which requires a license to practise automotive engineering. The State of Virginia has a law requiring the examination and the certification of architects and professional engineers, but the Secretary has written expressing doubt whether this law would be construed to cover automotive engineers. The State Board of Engineers of Oregon does not recognize automotive engineering as a branch of professional engineering at this time, according to advices from the Secretary of that State. The Secretary of State of Arizona informs us that there is a State law requiring the licensing of architects, engineers, land surveyors and assayers but there is some question about its applying to automotive engineers. A bill to repeal the licensing of engineers has passed the Senate in Pennsylvania and has gone to the third reading in the House.

The members must appreciate that it is a serious matter for the Society office to accept the responsibility of advising members whether they should register in their respective States or not. It would seem advisable for each of the members to write the Secretary of the State in which he resides or practises to secure an opinion or interpretation that carries some legal authority with it.

In response to the request contained in the circular letter sent out by the Society, letters continue to be received from the members expressing opinions on the advisability of laws requiring the registration or the licensing of engineers. Judging from the letters received, opinion is about evenly divided. It is noteworthy, however, that the majority of the engineers who have attained some prominence in the industry are opposed to this form of legislation.

Spark-Advance in Internal-Combustion Engines

By G. B. Upton[1]

Semi-Annual Meeting Paper

Illustrated with Charts

ALTHOUGH the proper timing of the spark is as essential as the spark itself and the electrical and mechanical devices for producing the spark have been many, little attention has been given to the study of spark-advance. An error in timing of ± 20 deg. in a low-compression engine, or of ± 15 deg. in most other engines, has been shown experimentally to cause a loss of 10 per cent from the best power and economy, provided other conditions remained the same. Hand or semi-automatic control can average hardly closer than ± 15 deg. to the correct advance because the speed and the load combinations are constantly changing on the road.

Two important phases mark the spark-advance problem. The practical question as to whether the requirements of optimum spark-advance at various combinations of load and speed are such as might be controlled automatically is apparently answered affirmatively; it can be satisfactorily represented by additive functions, one of speed only and one of load or intake-suction only. Hand adjustment would still be needed to take care of the difference between a clean engine and a dirty one or of a cold engine. When once an engine has been warmed up, however, automatic controls could maintain the proper spark-advance, thereby increasing in practice the power, flexibility and economy of the engine.

The second phase is that of scientific analysis. Inasmuch as combustion takes time, and as the engine is rotating during the combustion process, spark-advance is essential, for it maintains a definite relation between the progress of the explosion and the motion of the piston of the engine. This relation should be such that half the rise of pressure during combustion would occur at the dead-center position of the piston. Analysis, however, both theoretical and experimental, shows that one-half of the pressure rise occurs substantially at three-fourths of the explosion-time, that is, the interval between the point of ignition and the pressure peak. This, then, is the numerical basis for the relations of explosion-time, engine speed and optimum spark-advance.

The existing data, relating to the explosion-time as affected by the mixture-ratio, the size of the combustion-chamber, turbulence, dilution with dead or exhaust gases and the temperatures preceding the explosion, are reviewed. Density is shown not to affect the explosion-time. The factor commonly supposed to be density, which demands an increased spark-advance as the engine is throttled, is in reality dilution with exhaust gas, which increases as the throttle closes, and the cause of the faster explosion in a high-compression engine than in one of low compression is the temperature preceding ignition.

A simple mathematical law, connecting the explosion-rate and turbulence and derived from experiments on bombs, is shown to be applicable to engines, and the manner of its application to the turbulence factor of any engine is indicated. This opens the way to quantitative experiments on turbulence in various designs of engine, hence to the development of designs for producing the greatest amount of turbulence, if such development should seem desirable, as turbulence is the factor that makes really high rotative speeds compatible with a good power output. An equally important factor in making explosions in engines occur more quickly than those in bombs is the heat produced adiabatically during the compression stroke.

The slowing-up of combustion on account of the dilution of the charge with exhaust gas was measured experimentally, and the results are tabulated and compared with the numerical extent of the dilution. Assuming that the dilution ratio is the ratio of the total charge to the quantity of new gas, the slowing-up of combustion because of dilution is demonstrated experimentally to be about proportional to the cube of the mass-dilution ratio.

The scientifically valuable part of the paper is the naming of the factors that affect the explosion-time of an engine, the giving of mathematical expressions for their laws of action and the finding of the numerical values of constants for such factors as turbulence and dilution. The measurement of the optimum spark-advance is made available as a research method for investigating the reaction rates of combustion, and hence of all that related group of topics now of interest to automotive engineers.

By measuring the optimum spark-advance, the combustion rates of gasoline with and without "anti-knock," or tetraethyl-lead, were measured. Although the quantity used was 20 times the normal amount, no change in the reaction rate of combustion was found when the combustion remained normal, that is, without detonation. When detonation occurred without anti-knock, the reaction times with anti-knock followed those to be expected with normal combustion. Detonation apparently changed the combustion habit as if it produced an abnormal top to the combustion.

IN the literature of automotive engineering there is little to be found about spark-advance. There is plenty of material about the electrical and mechanical design and the operation of apparatus for producing the ignition spark but the proper timing of the spark seems to be assumed. Ignition without timing control is comparable logically to a carbureter that will produce merely some kind of a fuel-air mixture without proportioning the mixture to the load and the speed of the engine.

It can be shown experimentally that the timing tolerance within which the spark must occur to get within 10 per cent of the best power and economy of the engine is about ± 20 deg. in a low-compression or 4 to 1 engine, and about ± 15 deg. in other engines. It may well be assumed that the average regulation of the spark by an ordinary driver of a car is not nearly as close as this.

The problem of spark-advance has two phases of importance: (*a*) finding whether the requirements of the optimum spark-advance at various combinations of load

[1] M.S.A.E.—Professor of experimental engineering, Cornell University, Ithaca, N. Y.

and speed are such as might be met by simple automatic apparatus; and (*b*) investigating scientifically the underlying theory or principles concerned. To the practical question the answer apparently is yes; optimum spark-advance can be satisfactorily represented by additive functions, one of speed only, the other of load, or intake-suction, only; hence a simple mechanism can be made to cover both the load and the speed effects. The scientific investigation is of extreme interest, for it seems that determining the optimum spark-advance involves a direct measurement of the volumetric or the spacial flame-speed; and a study of the variation of the optimum spark-advance with the load or the intake suction, the speed and the mixture-ratio leads to a quantitative numerical evaluation of the reaction-rate of combustion, turbulence, dilution with spent gas, and the like; all of which make a very valuable auxiliary and independent check on such fundamental investigations as those recently published by Midgley and his associates.

Review of Data

The first stage of an engineering investigation is a survey of extant scientific data. Next come the comparing and the correlating of such data with empirical practice. Good collections of data for our purpose are to be found in A. W. Judge's book, Automobile and Aircraft Engines, and in vol. 1, on Mechanics and Heat, of Glazebrook's Dictionary of Applied Physics.

Research on "explosions" in bombs brings out this essential that the explosion-time, or the time-duration of an explosion, is the time elapsing between ignition and the pressure peak, or the time of maximum pv product, if the volume is changing. The explosion-time is found to be

(1) Dependent on the mixture-ratio and the fuel used
(2) Independent of the absolute pressure, or the density, before ignition

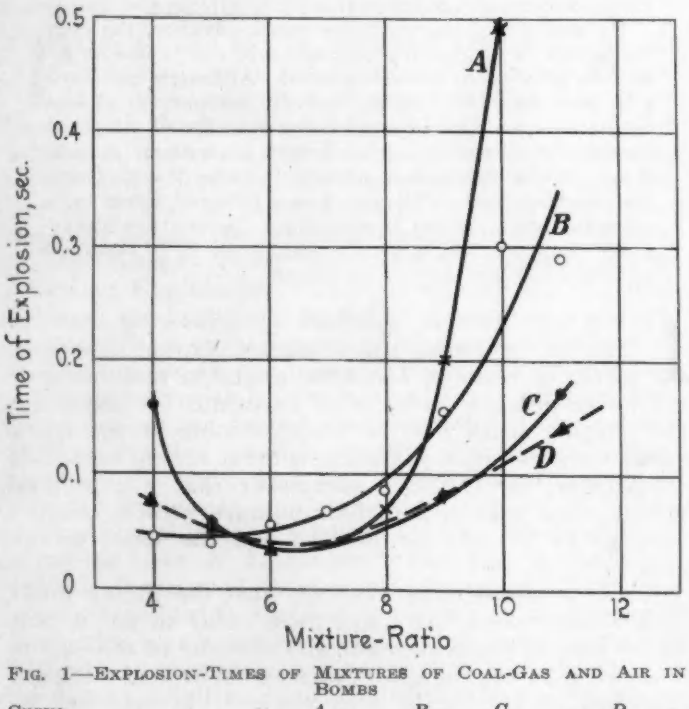

Fig. 1—Explosion-Times of Mixtures of Coal-Gas and Air in Bombs

Curve	A	B	C	D
Authority	Bairstow & Alexander	Dugald Clerk	Dugald Clerk	Massachusetts Institute of Technology
Diameter of Bomb, in.	10	7	7	....
Length of Bomb, in.	18	7	7	....
Initial Pressure, lb. per sq. in.	55	14.7	14.7	....
Kind of Gas	..	London	Oldham	Boston

(3) Somewhat dependent on the size and the shape of the explosion-chamber and the position of the ignition-point
(4) Greatly decreased by the turbulence of the mixture

The bomb experiments available in technical literature do not go into the effects on the duration of the explosion of two other variables, both of importance in an actual engine. These are

(1) Dilution of the charge with spent gases from previous explosions
(2) The temperature of the charge just before ignition

Experiments on an engine show that the mixture of maximum power for any given condition of speed and load is substantially identical with the mixture for the minimum value of the best spark-advance. Any richer or leaner mixture than that of maximum power needs more spark-advance than the maximum-power mixture. Except on a cold engine, however, the extra spark-advance for mixtures reasonably near to the maximum-power mixture is small; and this experimental fact makes it possible to eliminate the mixture-ratio variable from the general discussion, and to revert to it only as a minor factor in a complete solution of the problem of spark-advance.

Since the time-duration of the explosion is independent of the density of the charge, this possible variable disappears. The change of spark-advance required on account of throttling is not a consequence of the change of the charge density, but of the change of dilution with spent gas and of the temperature preceding the ignition. A high-compression engine, when compared with a similar low-compression engine, has a faster explosion, not by reason of a different density of the charge, but by reason of a different dilution of the charge and of a different temperature preceding the ignition.

The effects of the size and the shape of the explosion or the combustion-chamber and the position of the ignition-point and also of the mechanical and electrical lag in the ignition system, can be taken care of in a given engine by finding a characteristic or imaginary explosion-time for that combustion-chamber, as a bomb, for the fuel, the mixture and the temperature used but without any dilution or turbulence effects. The absence of turbulence corresponds, of course, to a zero engine-speed.

The effects of dilution and turbulence can readily be found by brake tests of the engine over a fair range of combinations of speed and load; the dilution can be measured from observations of the exhaust and the intake-manifold pressures and temperatures and a knowledge of the compression-ratio of the engine. The effect of the dilution with spent gases is to increase the characteristic explosion-time of the carburetion chamber by multiplying it by a factor fixed by the extent of the dilution. The effect of turbulence is to decrease the characteristic explosion-time by multiplying it by a simple inverse function of the engine speed. The actual explosion-time from ignition to the time of maximum pressure, or the pressure peak with the engine warmed-up to normal running conditions, can be given as the product of the three independent factors; (*a*) the "characteristic time" of the combustion-chamber as a bomb, (*b*) the dilution factor and (*c*) the turbulence or speed factor.

The actual explosion-time or the duration is simply related to the optimum angle of spark-advance and the rotation speed of the engine. If R is the speed of the engine in revolutions per minute, its angular speed is $360R/60 = 6R$ deg. per sec. When a is equal to the

angle of spark-advance, the timing of the spark ahead of the dead-center position of the engine, in seconds, is $a/6R$. It can be shown that for optimum spark-setting $a/6R$ = ¾ × the actual explosion-time = ¾ × the "characteristic time" × the dilution factor × the turbulence or speed factor. In this relation the characteristic time is a constant for a given engine and its ignition system under the ordinary operating conditions of the engine.

Explosion-Time

I shall review briefly the literature on explosion-time. Fig. 1 summarizes tests by various experimenters with mixtures of coal-gas and air. It will be seen that for mixture-ratios from 5.0 to 7.5 by volume, the explosion-time runs from 0.04 to 0.06 sec. with relatively minor effects on account of the size of the bomb, and none from the change of initial pressure or density. Fig. 2 shows

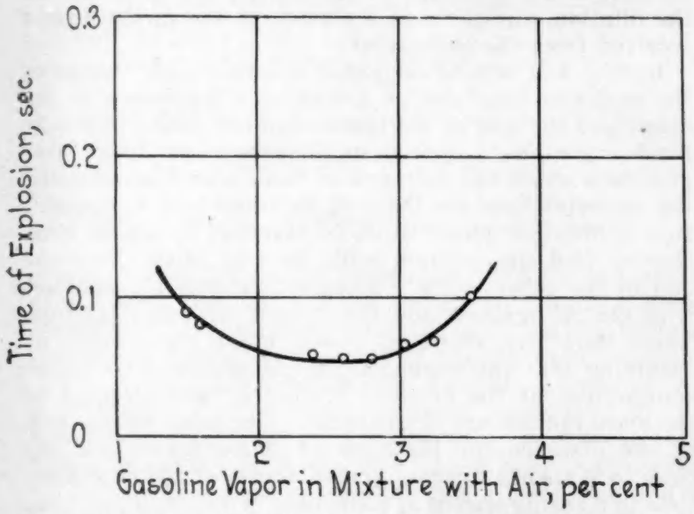

Fig. 2—Results of Experiment at Massachusetts Institute of Technology with a Gasoline Vapor-Air Mixture

from results obtained at the Massachusetts Institute of Technology that mixtures of gasoline vapor and air have the same characteristics as those of coal-gas and air; the minimum explosion-time in the bomb was 0.058 sec. In contrast to this explosion-time for gasoline in the bomb, Watson[2] gives the explosion-time in a racing-engine as 0.0055 sec. with single-spark ignition, and 0.0037 sec. with double-spark ignition. The speeding-up of the explosion in the engine, more than 10 times as compared with that of the bomb, is said to be due to the turbulence effect in the engine. Humphrey[3] found the explosion-time to be 0.026 sec. in a 28½ x 30-in. cylinder at 128 r.p.m.; Clark[3] found the explosion-time in a 9 x 17-in. cylinder at 180 r.p.m. to change from 0.033 sec. under normal operation to 0.080 and 0.090 sec. when the turbulence was allowed to die out before ignition.

Judge[4], by pressure-time diagrams, shows that the explosion-time in a certain engine was 0.013 sec. with gasoline as fuel, and 0.014 sec. with benzol. Comparing the data as to the explosion-times of various fuels[5] tested under like conditions, it seems that changing the fuel has a very small effect on the explosion-time. Alcohol, however, might be different from the hydrocarbon fuels,

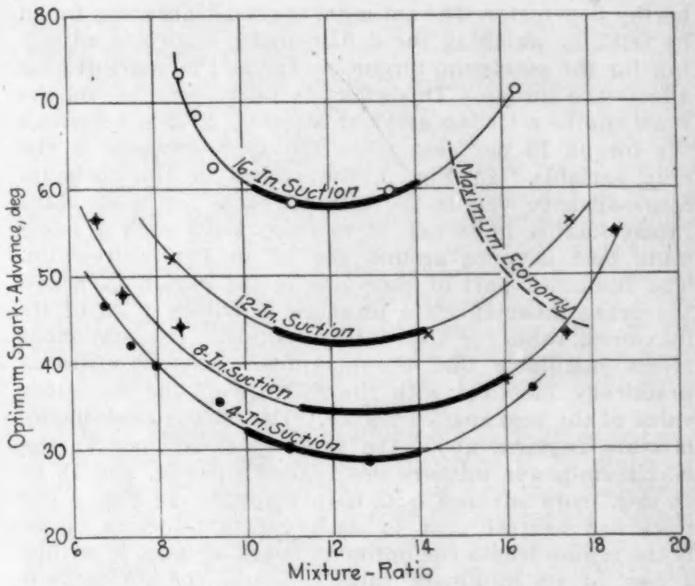

Fig. 3—Results of Optimum Spark-Advance Tests on a Ford Engine

which, having substantially the same explosion-times, could be substituted for one another in an engine without any appreciable change of the optimum spark-advance.

It will be noted that the curves of Fig. 1 of the explosion-time plotted against the mixture-ratio are fairly flat over a considerable range of mixture-ratios. This suggests that the optimum spark-advance in the practical operation of an engine may not be radically affected by the mixture-ratio. Tests on this point were first made by us on a Ford engine in the Cornell University laboratories with the results shown in Figs. 3 and 4. The airflow to the carbureter was measured by a Durley meter and box and the gasoline flow by a modified Pen-

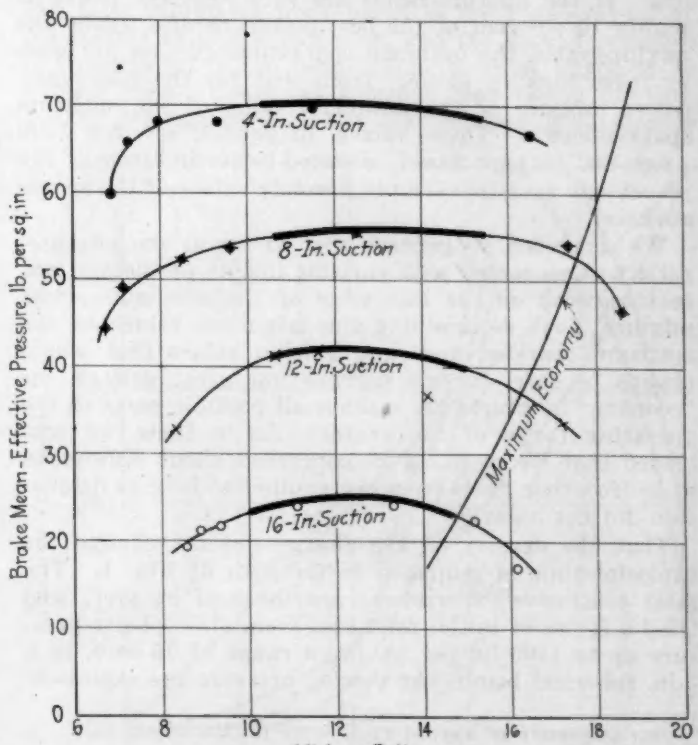

Fig. 4—Curves of the Brake Mean-Effective Pressure Developed by a Ford Engine at Various Mixture-Ratios

[2] See Dictionary of Applied Physics, by R. Glazebrook, vol. 1, p. 299.

[3] See Dictionary of Applied Physics, by R. Glazebrook, vol. 1, p. 300.

[4] See Automobile and Aircraft Engines, by A. W. Judge, p. 30.

[5] See the curves of results obtained at the Massachusetts Institute of Technology with coal-gas and air in Fig. 1 and gasoline and air in Fig. 2.

berthy flow-meter. The optimum spark-advance was found by trial, by watching the dynamometer scale and adjusting for the maximum torque, so far as the spark-timing affects the torque. This effect is fairly definite; on the Ford engine a timing error of about ± 20 deg. decreases the torque 10 per cent when the spark-advance is the only variable. In Fig. 3, the curves of the optimum spark-advance versus the mixture-ratio for a constant intake-suction in inches of mercury have each a minimum best advance around the 12 to 1 mixture-ratio. The thickened part of each line is the region in which the brake mean-effective pressure is within 1 lb. of its maximum value for that intake-suction. The statement seems justifiable that the maximum-power mixture is practically identical with the mixture of the minimum value of the best spark-advance. The maximum-economy mixture requires about the same spark-advance as the maximum-power mixture near closed throttle, and 10 to 15 deg. more advance near open throttle. In Fig. 4 the thickened central part of each constant-suction curve is the region where the optimum spark-advance is within 2 deg. of its minimum value. Again, the comparison shows the practical identity of the mixtures for the maximum power and for the minimum best spark-advance. The conclusion is, of course, that the maximum-power mixture is the fastest burning mixture of the series, or the mixture of the highest reaction-velocity.

The same general conclusions were arrived at from tests on a Continental 7-R six-cylinder engine in the same laboratory. The test procedure was here more elaborate; at each of the six fuel-flow rates five or more tests were made of the brake mean-effective pressure versus the spark-advance. From the curves of these tests the spark-advances for the best mean-effective pressures were picked off, with the corresponding values of the mean-effective pressures so as to plot the curves in Fig. 5. The tolerance in the spark-advance in this engine to keep within 10 per cent of the best power, is less than ± 15 deg. If the mixture-ratios are such that the power is within 10 per cent of the best power, as affected by the mixture-ratio, the optimum spark-advance does not vary by more than ± 10 deg. from that for the maximum-power mixture or the minimum value of the optimum spark-advance. These values, of course, are for 1000 r.p.m. and perhaps would be stated better in terms of the percentage variation than in absolute values of the spark-advance.

We concluded, in general, that so far as the mixture-ratio was concerned as a variable in this problem it was best to work on the lean edge of the maximum-power mixture, thus determining the minimum values of the optimum spark-advance and getting values that would always be safe, though perhaps not great enough for "economy" mixtures throughout all possible parts of the operating range of the engine. As to fuels, we concluded that we need not be concerned about differences of hydrocarbon fuels from one another so long as detonation did not occur.

That the density of the charge did not change the explosion-time is indicated in the data of Fig. 1. The most conclusive experiments are those of Petavel[6], who tried a series of initial pressures from atmospheric pressure up to 1100 lb. per sq. in., a range of 75 to 1, in a 4-in. spherical bomb. At the top pressure the explosion-time of a 1 to 6 volumetric mixture of coal-gas and air was still 0.058 sec., the same, within the accuracy of measurement, as at the low pressures. The density of the charge, then, is not a factor in this problem. The same conclusion follows from Midgley and Janeway's recent paper[7] on Laws Governing Gaseous Detonation. They found that mass rate of "burn" was proportional to the first power of the initial density. Since mass is volume times density, it follows that the volumetric or the space rate of flame travel is independent of the initial density. Engines of similar design but of different compression-ratios have different combustion rates, not because of the different densities at ignition, but because of the different temperatures preceding combustion and the different dilutions with dead gas, and perhaps because of differences in turbulence. Also, in throttling an engine the operative factor calling for more spark-advance is not a change of the density but a change of the dilution, and perhaps a change of the mixture-ratio received from the carbureter.

In Fig. 1 it will be seen that no systematic change in the explosion-time can be traced to a difference in the shape and the size of the bomb chamber used. The only bomb experiments quoted in Glazebrook or Judge that give data about the distance of flame-travel as affecting the explosion-time are those of Bairstow and Alexander[8]. In a cylindrical bomb, 10 in. in diameter by 18 in. long, they shifted the ignition-point by 3-in. steps from one end to the other. The explosion-time was the shortest with central ignition and the longest with end ignition. Their data are reasonably well fitted empirically by assuming that the explosion-time is proportional to the square root of the distance from the ignition-point to the more remote end of the bomb. The same square root of the distance for the time of flame travel also fits close to Watson's figures,[8] quoted above, of the explosion-time in a racing engine at 0.0055 sec. with single ignition, and 0.0037 sec. with double ignition. It seems probable that in an actual engine the shape of the combustion-chamber and the position of the spark-plug or plugs in it will be more important practically in determining the explosion-time than the absolute dimensions of the combustion-chamber.

Data on the effect of the initial temperature on the explosion-time in bomb experiments seem quite lacking. The best information seems to be given in Midgley and Janeway's paper[7] on the Laws Governing Gaseous Detonation, where it is found that the combustion rate varies approximately as the cube of the absolute temperature before ignition. For a given engine the rise of the temperature during compression is a definite multiple of the temperature at the beginning of compression; this in turn is fixed largely by the temperature of the air entering the carbureter. Hence, for a given engine the explosion-time and the spark-advance vary inversely, by some power higher than 1, as the absolute temperature of the entering air. In comparing one engine with another, since the rise of the temperature during compression varies with the y-1 power of the compression-ratio, and the combustion-rate varies with the cube of the absolute temperature preceding ignition, we shall have a combustion-rate varying as the compression-ratio to the $3(y\text{-}1) = 3(1.33 - 1) = \pm 1.0$ power; in other words, in engines similar but of different compression-ratios the optimum spark-advance should vary inversely as the compression-ratios.

The increase of the combustion-rate with the cube of the absolute temperature may well be the reason that, in constant-volume combustion, the combustion-rate in-

[6] See Dictionary of Applied Physics, by R. Glazebrook, vol. 1, p. 298.

[7] See THE JOURNAL, April, 1923, p. 367.

[8] See Dictionary of Applied Physics, by R. Glazebrook, vol. 1, p. 299.

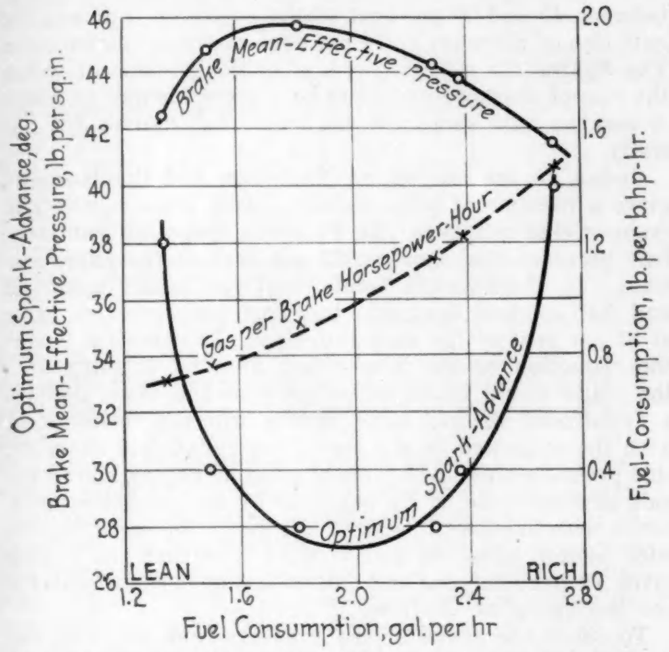

FIG. 5—OPTIMUM SPARK-ADVANCE AND BRAKE MEAN-EFFECTIVE PRESSURE CURVES FOR A CONTINENTAL SIX-CYLINDER ENGINE

creases as the combustion progresses. For the expansion of the first portions burned causes an adiabatic compression of the unburned residue and the consequent rise of the temperature ahead of the chemical reaction.

The lesser spark-advance that satisfies an engine, as it progresses from clean to dirty, or "carboned," in the combustion-chamber, is also largely a temperature effect. A dirty engine has less loss of heat to the walls of the combustion-chamber than has a clean one and a higher temperature of the charge at the end of compression; hence it needs a decreased spark-advance.

Analysis of Shape of Pressure-Time Curves

The fact that the explosion-time varies as about the square root of the distance the flame has to travel suggests a simple line of attack on the analytical problem of the shape of the pressure-time curves of the explosions. In turn, the shape of the pressure-time curve determines what fraction of the explosion-time should come ahead of the dead-center position of the piston. This fraction of the explosion-time ahead of dead-center is the tying relation between the explosion-time and the optimum spark-advance. Too early ignition leads to losses through the compression of the burnt charge after combustion by the motion of the piston; losses partly from the excessive heat-transfer to the combustion-chamber walls, and partly from abnormally large dissociation effects stopping combustion. Too late ignition means that combustion is still going on appreciably as the expansion stroke begins, and hence leads into losses best understood from the thermodynamic theory that heat is less available for conversion into work when it is released at lower temperatures. Since the piston actually does move while combustion is going on, instead of being instantaneous as is postulated in the thermodynamic cycles in the textbooks, the actual timing of the combustion process in the engine cycle is a compromise between too early and too late and involves elements of both. Combustion should begin while the piston is moving up, and should end while the piston is moving down. Just where, with regard to the piston motion, should the combustion be timed for best efficiency?

If the time of flame travel in constant-volume combustion is proportional to the square root of the distance, the velocity of the flame travel will vary as the first power of the *time* from ignition. This is not in conflict with Midgley's calculations,[9] for he has been studying the combustion-rate as a function of the pressure and the temperature, not as a function of the time from ignition. Furthermore, flame velocity is actually a compound affair; the flame travel has two components, one of actual velocity with regard to the gas, and the other of the mass-motion of the gas itself. No attempt will be made in this paper to separate these two components of flame velocity with regard to the cylinder.

I have assumed a combustion-chamber of cylindrical form, with a height equal to one-third of its diameter, and with the ignition-point in one end, half-way from one side to the center. The time for a complete combustion, the explosion-time, I have taken as unity. The final velocity also has been taken as unity. With the velocity proportional to the time, the same decimal then stands for the time or for the velocity, in terms of their terminal values. The distance of the flame-front from the ignition-point varies as the square of the time from ignition. Up to about one-half of the time of complete combustion the flame-front makes a hemispherical surface about the ignition-point, the flame having not yet been cut-off by running into the combustion-chamber walls. In this first half of the combustion-time, therefore, the area of the flame-front varies as the square of the distance of the flame-front from the ignition-point and, in consequence, as the fourth power of the time from ignition, since the distance itself varies as the square of the time. The volumetric rate of combustion is the product of the velocity of the flame and the area of the flame-front; hence the volumetric rate of combustion varies as the fifth power of the time from ignition. The total volume or mass burnt is the time-integral of the volume rate of combustion, and the rise of pressure is proportional to the total volume or mass burnt up to a given time. Hence, we conclude that during the first half of a combustion that is started from a spark-plug

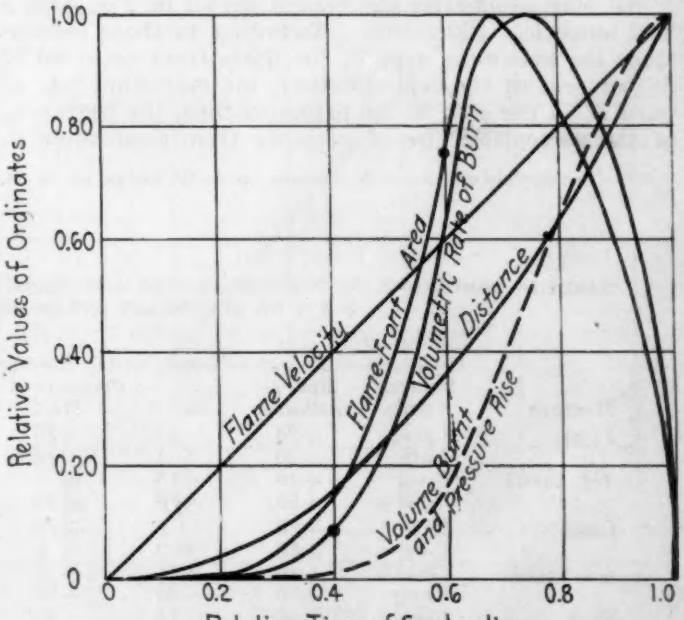

FIG. 6—THEORETICAL STUDY OF THE PROGRESS OF COMBUSTION IN A CYLINDRICAL COMBUSTION-CHAMBER

[9] See THE JOURNAL, April, 1923, p. 367.

in a cylinder wall, the rise of pressure will vary approximately as the sixth power of the time from ignition.

This is the explanation of the shape of the beginning of the pressure-time curves of combustions in either bombs or engine cylinders. On such curves there appears to be an appreciable "lag," or time-interval, in which there is no perceptible rise of pressure. On a sixth-power law, the pressure at the time ½ is 1/64 of that at the time unity, or simply too small to notice, relatively. Fig. 6 shows curves representing the mathematical relations outlined above.

At about one-half the time of a complete combustion the flame-front ceases to be hemispherical, being cut-off in part by running into the walls of the combustion-chamber. From this time on the flame-front begins to lose area, first slowly, then rapidly, and becomes zero as the combustion ends. The exact law of the change of flame-front area with position or time depends on the shape of the combustion-chamber and the position of the ignition-point, what we may call the geometric lay-out in our case. For the simple combustion-chamber chosen and described I have worked out these flame-front areas for the last half of the combustion-time; and they are plotted in Fig. 6 with an arbitrary choice of a maximum area of flame-front equal to unity.

The product of the flame velocity and the flame-front area gives the volumetric rate of "burn." This continues to rise for a little while even after the flame-front area has begun to decrease, because the flame velocity is still increasing; but the decreasing area of the flame-front soon becomes the dominant factor, and the volumetric rate of burn must reduce to zero as the combustion ends.

The time integral of the volumetric rate of burn is the volume burnt and is proportional to the rise of the pressure from the combustion. This calculated curve of pressure-rise plotted against time in Fig. 6 is the end toward which we have been working. It is shown to be a direct consequence, in its form, of the law of flame velocity plotted against time and the geometry of the combustion-chamber. The velocity-time law chosen was the simplest possible; it remains to be seen whether the results deduced from it give a fair picture of the actual progress of combustion in bombs or engine cylinders.

We may summarize the results shown in Fig. 6 in a few numerical statements. According to these calculations the maximum area of the flame-front occurred at 68 per cent of the explosion-time, the maximum rate of burn at 74 per cent of the explosion-time, the beginning of the perceptible rise of pressure from combustion at between 40 and 50 per cent of the explosion-time and the half rise of pressure at 74 per cent of the explosion-time. The figures for the fractions of explosion-time at which the rise of pressure begins to be perceptible and at which it reaches half value are the important figures for our study.

Judge, in the chapter on Explosion and Combustion[10], gives a number of pressure-time cards from bombs and from engine cylinders. In Petavel's spherical bomb, the half pressure-rise came at 85 per cent of the explosion-time. In Hopkinson's bomb that was nearly spherical and had central ignition, the half pressure-rise came at 81 per cent of the explosion-time. In spherical bombs the reduction of the flame-front area by running into the walls would be expected later in the burn than in a cylindrical vessel; these figures are not inconsistent with the computed 74 per cent for a cylindrical chamber. One pressure-time card from a gasoline engine shows the half pressure-rise at 65 per cent of the explosion-time, while with another it is at 72 per cent; the same engine, with benzol, gives 69 per cent; a producer-gas engine gives 79 per cent for the half rise, and 44 per cent for the beginning of the rise.

To check the shape of the pressure-time curve in our own experiments we used a 7-hp. stationary Fairbanks hit-or-miss engine in the Cornell University laboratory. The speed of 340 r.p.m. made it possible to take indicator-cards with a Crosby indicator; the indicator-drum drive was offset 90 deg. so that pressure-time cards were obtained. In this engine we found the ratios to the total explosion-time for the beginning of the rise of pressure and for the half rise to be 0.54 and 0.81 on a lean mixture; on the maximum-power mixture, 0.54 and 0.77, and on a rich mixture, 0.44 and 0.71. Altogether, it seems that, for engines in general, we may assume pretty accurately that half pressure-rise will occur at, or very close to, 75 per cent of the explosion-time. The theoretical conclusions of Fig. 6 are confirmed experimentally, though the assumption as to the law of flame velocity was so inadequately simple.

Correlation of Optimum Spark-Advance and Explosion-Time

This brings us to the relation, or correlation, between the optimum spark-advance and the explosion-time. For we may reasonably assume that we wish to time the spark, for best results, so that the combustion will occur at one-half before and one-half after the dead-center position of the piston, or that the half pressure-rise will occur at the dead-center. This makes about the best approximation to a vertical combustion line on the pressure-

[10] See Automobile and Aircraft Engines, by A. W. Judge, pp. 28-30, 32, 55-56.

TABLE 1—COMPARISON OF PRESSURE-VOLUME AND PRESSURE-TIME INDICATOR-CARDS OBTAINED FROM FAIRBANKS 7-HP. 5 X 7 IN. STATIONARY HIT-OR-MISS ENGINE, RUNNING AT 340 R.P.M.

Mixture	Timing by Pressure-Volume Cards	Pressure-Time Cards; Timing, Deg. of Spark-Advance	Pressure-Rise Begin	Pressure-Rise Half	Pressure-Rise Peak	Explosion-Time, Sec.	Ratio to Explosion-Time: Beginning of Pressure-Rise	Ratio to Explosion-Time: Half Pressure-Rise
Lean	Later	+20	−15	−25	−40	0.029	0.58	0.75
	Late	+30	− 1	−14	−22	0.026	0.60	0.85
(½ turn)	Good	+40	+15	− 1	− 9	0.024	0.51	0.84
	Good+	+50	+29	+13	+ 3	0.023	0.45	0.79
Good	Late	+20	− 8	−20	−31	0.025	0.55	0.78
	Good−	+30	+ 2	− 8	−23	0.026	0.53	0.72
(⅞ turn)	Good+	+40	+11	+ 1	− 9	0.024	0.59	0.80
	Early	+50	+25	+10	+ 2	0.025	0.48	0.77
Rich	Late	+20	−15	−25	−40	0.029	0.58	0.75
(1½ turns)	Good−	+30	+ 4	−12	−30	0.029	0.43	0.70
	Good	+40	+15	− 1	−22	0.030	0.40	0.66
	Early	+50	+28	+ 4	−14	0.031	0.34	0.72

TABLE 2—AVERAGE RESULTS FROM FAIRBANKS ENGINE TESTS

Mixture	Explosion-Time, Sec.	Fraction of Explosion-Time to Pressure: Begin	Half	Ignition to Beginning of Pressure-Rise	Actual Time, Sec.: First-Half of Pressure-Rise	Last-Half of Pressure-Rise
Lean	0.0255	0.54	0.81	0.0138	0.0069	0.0048
Good	0.0250	0.54	0.77	0.0135	0.0058	0.0058
Rich	0.0299	0.44	0.71	0.0132	0.0081	0.0087
Average of all Mixtures	0.0268	0.52	0.76	0.1350	0.0070	0.0064

Calculated optimum spark-advance, for half pressure-rise exactly at dead-center; lean mixture, 42 deg.; good mixture, 39 deg.; rich mixture, 43 deg.

volume indicator-card, and we know pretty well from practice that a nearly vertical combustion line in that card is indicative of the best power and efficiency of the engine.

To check the assumption that the timing of the spark so that the half pressure-rise will occur at the dead-center, or that the explosion-time will be three-quarters over at dead-center, for the best pressure-volume indicator-card shape, we made experiments on a 7-hp. Fairbanks stationary engine. Both pressure-volume and pressure-time indicator-cards were taken. The results are given in Table 1. The first column of the table gives the mixtures; the measurement was in turns of a needle-valve in a plain venturi carbureter. The second column gives the judgment as to the spark-timing from the shape of the pressure-volume indicator-card. Late timing is indicated by — signs and early timing by + signs; the measurements of the timing are in degrees of crankshaft rotation from the dead-center. The spark-advance was measured on a calibrated index-scale on the timer by jump-spark ignition. Particular attention is called to the correlation of the second column, the timing by the pressure-volume card with the fifth column, the timing of the half pressure-rise in crank-angle degrees. It will be seen that the cards called "good" in pressure-volume card shape had the half pressure-rise substantially at the dead-center.

Table 2 summarizes and averages the results of the tests in Table 1. Each result in Table 1 is itself the average of four or five cards of each kind, either of pressure-volume or of pressure-time. It is interesting to study, in Table 2, the change in the combustion habit with the mixture-ratio. The time from the ignition to the beginning of the pressure-rise seems to be independent of the mixture-ratio. The first half of the rise of pressure of the lean mixture is slow and the last half fast, as compared with the other mixtures. The rich mixture is slow in both halves of the pressure-rise but more so in the second half. The maximum-power mixture is not only the quickest to burn, but has a considerably more rapid and more symmetrical pressure-rise. The very high speed of burning in the last half of the pressure-rise of the lean mixture may be tied up with the fact that such mixtures are the only kind that backfire in an engine. The calculated optimum spark-advances for all three mixtures in Table 2 are astonishingly close, 42, 39 and 43 deg., and these results confirm our tests on the Ford and Continental engines very nicely. They also check the 40-deg. spark-advance data in Table 1 closely. The optimum spark-advance in Table 2 is calculated by converting the time in seconds for each mixture, from the ignition to the half pressure-rise, into crankshaft degrees ahead of the dead-center.

We may consider that we have proved experimentally what we deduced theoretically, that the optimum spark-advance is such that the half pressure-rise occurs at the dead-center; and that this stage of the pressure-rise occurs practically at 75 per cent of the explosion-time after ignition. Writing a for the spark-advance in degrees, and a_0 for the optimum advance, e for the explosion-time, R for the number of revolutions per minute, we have then $a/6R =$ the spark-advance in seconds, and $a_0/6R = \frac{3}{4}e$ whence $a_0 = 9eR/2$ and $e = 2a_0/9R$. By finding a_0 for an engine on the test stand, through the change of the spark as a single variable, noting the torques resulting while the speed, the intake-suction, the temperatures and the like are held constant, we can measure e, the explosion-time, under the conditions of the test. So through the optimum spark-advance we can investigate changes of the explosion-time by mixture variation, dilution, turbulence and the like and open the door to a wide field of study of combustion processes in engines.

TURBULENCE

I have mentioned already that turbulence is a factor in speeding-up combustion, quoting some results in engine tests by Dugald Clerk. Hopkinson[11] experimented on turbulence with a fan inside a cylindrical bomb 12 in. in diameter. He used a 1 to 9 mixture by volume of coal-gas and air. The data of his tests are plotted in Fig. 7. Assuming, as a most simple relation, that the speed of the flame travel is a linear function of turbulence, we should have flame speed equals the size of the bomb divided by the explosion-time or $K\ (1 + bR)$, where K is the reciprocal of the explosion-time without turbulence, and R is the revolution speed that measures the turbulence. Fig. 7 shows that this simple equation is satisfactory for the bomb tests. Later, it will be shown that the same type of equation holds good for an engine; R is then the speed of the engine in revolutions per minute. The constant b we shall call the turbulence factor. For Hopkinson's fan and bomb, the constant b

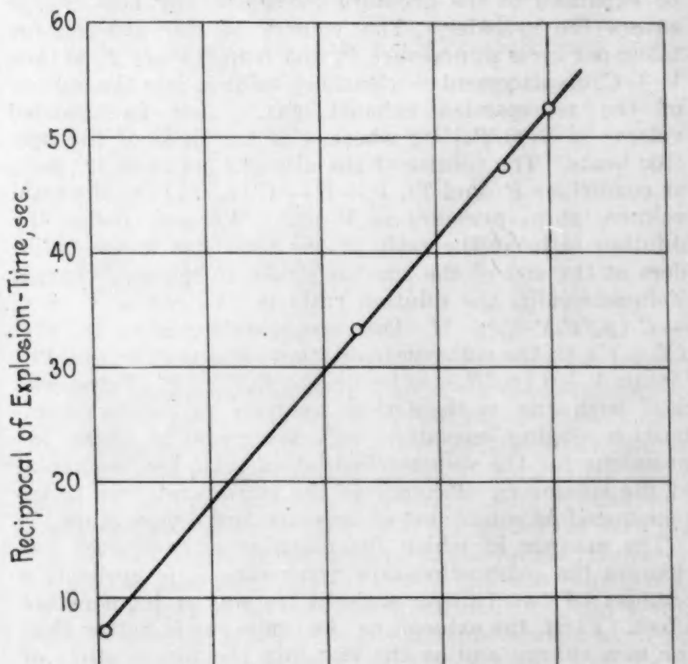

FIG. 7—RESULTS OF HOPKINSON'S EXPERIMENTS WITH TURBULENCE FROM A FAN IN A BOMB

[11] See Automobile and Aircraft Engines, by A. W. Judge, p. 55.

is 0.0016; in automotive engines b becomes of the general magnitude of 0.001. This means that turbulence cuts the explosion-time in two at 1000 r.p.m., divides it by 3 at 2000 r.p.m. and so on. In racing-engines the explosion-times drop to 1/5, and in engines with small cylinders they drop perhaps to 1/10.

In finding the turbulence factor for a given engine one must first eliminate from the results the dilution factor, the slowing-up of combustion by dilution with exhaust gases that have been caught in the clearance volume. The dilution at a constant speed increases as the throttle is closed, but is not zero at full throttle. After the dilution effect has been eliminated from the test results, the curve of the optimum spark-advance for zero dilution versus the speed can be analyzed for the turbulence factor after the manner of Fig. 7. Since e, the explosion-time, $= 2a_0/9R$, the reciprocal of e is $9R/2a_0$. A plot of $9R/2a_0$ for zero dilution against R will find the constants $1/e_c$ where e_c is the characteristic explosion-time of the combustion-chamber of the engine as a bomb without turbulence or dilution, and b, the turbulence factor of the engine.

Dilution with Exhaust Gases

Dilution with exhaust gases is never absent in Otto-cycle-engine operation. At the end of the exhaust stroke of a 4-stroke-cycle engine the clearance volume C of a cylinder is filled with exhaust gas at an absolute pressure of p_e and an absolute temperature of T_e. As the piston moves down on the suction stroke this exhaust gas, or the spent or dead gas, re-expands behind the piston. It continues to expand until the pressure drops to P_i, the intake-manifold pressure. Of course, when an engine with mechanically operated rather than automatic intake-valves is throttled, the exhaust gas flows from a cylinder into the intake-manifold when the intake-valve opens and is later sucked back through the valve into the cylinder. Essentially, however, the clearance gas must be expanded to the pressure P_i before any new charge enters the cylinder. The volume of the new charge taken per cycle at pressure P_i and temperature T_i is then $V + C$, displacement + clearance volume, less the volume of the re-expanded exhaust gas. This re-expanded volume is $C(p_e/P_i)^{1/y}$; where y is the ratio of the specific heats. The volume of the new gas per cycle is, then, at conditions P_i and T_i, $V + C - C\,(p_e/P_i)^{1/y}$; the total volume, at p_i pressure is $V + C$. We may define the dilution ratio as the ratio of the total gas in the cylinders at the end of the suction stroke to the new charge. Volumetrically, the dilution ratio is $(V + C)/[V + C - C\,(p_e/P_i)^{1/y}]$. If the compression-ratio is $r = (V + C)/C$, the volumetric-dilution ratio is $r/[r/(p_e/P_i)^{1/y}]$ or $1 + [(p_e/P_i)^{1/y}/\{r - (p_e/P_i)^{1/y}\}]$. Those who deal with the mathematical analysis of internal-combustion engine operation will recognize in these expressions for the volumetric-dilution ratio the reciprocal of the volumetric efficiency of the engine referred to intake-manifold conditions of pressure and temperature.

The manner in which the dilution with exhaust gas changes the combustion-rate by slowing it is probably a complex of two things, each in its way a temperature effect. First, the exhaust or clearance gas is hotter than the new charge and as the two mix the temperature of the new charge is raised. This action does not disturb the volumetric calculations just given, because the expansion of one part of the gas by heating is counterbalanced by the contraction of another part by cooling. But the new charge, being hotter before ignition, might be expected to burn *more* rapidly because of the dilution. This does not happen, as it is swamped out and overcome by the second action of the dilution. This is the absorbing of the heat of combustion by the dead gases as the combustion progresses. The temperature to which the combustion would go if the gases were not diluted is greatly lowered by dilution. The operative factor is really the "heat-mass" of the diluting gases, their capacity for absorbing heat during the temperature rise, or the product of their actual masses by their specific-heats. As the new gas during combustion becomes exhaust gas, the specific-heats after combustion will be the same for the products from the new charge and for the products of a previous combustion that dilute the new charge. Hence, the dilution factor, which is operative in holding-down the combustion temperature, is simply the mass-dilution ratio of the total weight of the gas in the cylinder to the weight of the new charge. The heating of the new charge, previous to ignition, on account of its mixing with the diluting dead gases, is also nearly proportional to the mass-dilution ratio, the temperature being shifted from T_i toward T_e very nearly in the ratio of the masses of the old and of the new charges. This shift is affected somewhat by the fact that the specific-heats at temperatures T_i and T_e are not exactly the same; but the losses of heat to the cylinder walls during the suction and the compression strokes are much more important than any effect caused by the variation of the specific-heat with the temperature.

On the whole, then, we may expect the change in the combustion rate consequent on the dilution with dead gas to be some function probably of the mass-dilution ratio of the total gas to the new charge. And since the temperature effects are proportional to the mass-dilution ratio, and the temperature effects on the combustion rates can be represented fairly well by exponential functions, mathematically, we shall attempt to analyze the dilution effects by plotting the spark-advance versus the dilution ratios on logarithmic cross-section paper.

The volume $V + C - C\,(p_e/P_i)^{1/y}$ of the new charge, at temperature T_i and pressure P_i is mixed with a volume $C\,(p_e/P_i)^{1/y}$ of dead gas, of pressure P_i and temperature $T_e/(p_e/P_i)^{1/y}$. Since the density of gases at the same pressure is inversely proportional to their absolute temperatures, the mass-dilution ratio of the total gas to the new charge is $1 + (p_e/P_i)\ (T_i/T_e)/\{r - (p_e/P_i)^{1/y}\}$, an expression that may be compared with the corresponding one for the volumetric-dilution ratio.

In these expressions for volumetric or mass-dilution ratios the value of y, the specific-heat ratio, was left unspecified. We may choose two cases to be tried out on test results. First, because of its simplicity, we may see what will happen if $y = 1$, or the expansion is isothermal. Then the volumetric-dilution ratio becomes $1 + (p_e/P_i)/(r - p_e/P_i)$; and the weight-dilution ratio becomes $1 + (p_e/P_i)\ (T_i/T_e)/(r - p_e/P_i)$. The two expressions differ in this case only in the fact that the exhaust temperature is higher than the intake temperature. It follows that mass dilution is less than volumetric dilution under the same conditions.

There is another peculiar angle to this affair which is interesting. If T_e rises, at constant p_e and P_i, the cylinder gas does not get hotter, because the mass of the exhaust gases retained in the clearance drops as T_e increases in inverse proportion to T_e while its heat-content per unit mass increases in direct proportion to T_e. Hence, the heat added by the clearance gas to the new charge at constant p_e and P_i is fairly independent of T_e.

Actually we can hardly assume that the re-expansion

of the clearance gases is isothermal. If it is adiabatic, the value of y is around 1.30 to 1.35 and it will be convenient to take $1/y$ as 0.75. The analysis of the engine tests justifies this, as will soon be shown. Also, in determining mass dilution, T_e and T_i need to be measured while obtaining the experimental data.

In a wider field the change of the mixture-ratio may itself be taken as one kind of dilution. We may consider the perfect mixture, or the mixture for complete combustion, to be diluted either with excess fuel or with excess air. Actually this assumption is a rather poor one, in the case of rich mixtures at least, because the chemical reactions of combustion are not the same when the mixture-ratios are varied. But the nitrogen-component of the air probably acts as a diluent much as exhaust gas does in the engine. An inspection of the curves of explosion-time versus mixture-ratio, as in Figs. 1 and 2, shows a general tendency to a curve of cubic form, or thereabout, symmetrical about a vertex for the mixture at the maximum combustion-rate. It will be shown from the engine tests that combustion is slowed by dilution with exhaust gases in proportion to the inverse cube of the mass-dilution ratio.

In finding the dilution factor from engine tests one need not be particular about the value of p_e. At full load and speed, p_e may be appreciably higher than atmospheric pressure; but P_i is then close to atmospheric pressure, and dilution does not vary rapidly with p_e/P_i when this ratio is near unity. At light loads, p_e becomes substantially identical with atmospheric pressure; and P_i is equal to the atmospheric pressure less the intake suction. If P_a represents atmospheric pressure and S the intake suction, all pressures being in same units, of course, then p_e/P_i may be replaced by $P_a/(P_a - S)$ without much error. It is better, naturally, to measure values of p_e and P_i; but many extant data can be made available for fairly accurate determinations of dilution from the recorded values of S, with p_e unknown and P_a found by a reasonable guess.

Original Cornell University Tests

Our first tests to check the theory we have outlined, as affecting the optimum spark-advance for various loads and speeds, were made on a Ford engine in the Cornell University laboratory. This is a stock engine with the exception of minor changes; it has Dow metal pistons, an Atwater Kent open-circuit ignition-system, and a Rayfield thermostat to control the temperature of the discharge water. The carbureter is of the regular Holley type. Instead of the air-horn for supplying heated intake-air we used electric heating, and the air was metered by a Durley orifice. The temperature of the entering air was kept at 140 deg. fahr. unless otherwise noted; the discharge water also was at 140 deg. fahr. A Froude water-brake was used to measure and absorb the torque. Gasoline was measured by a specially calibrated Penberthy flow-meter.

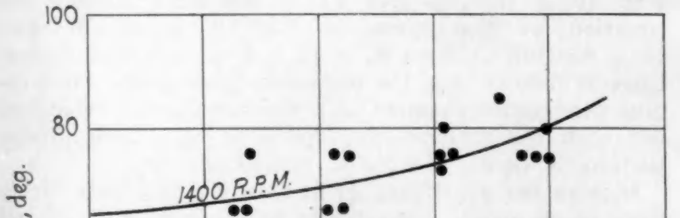

FIG. 8—EFFECT OF VARYING THE INTAKE SUCTION ON THE OPTIMUM SPARK-ADVANCE OF A FORD ENGINE AT A CONSTANT SPEED

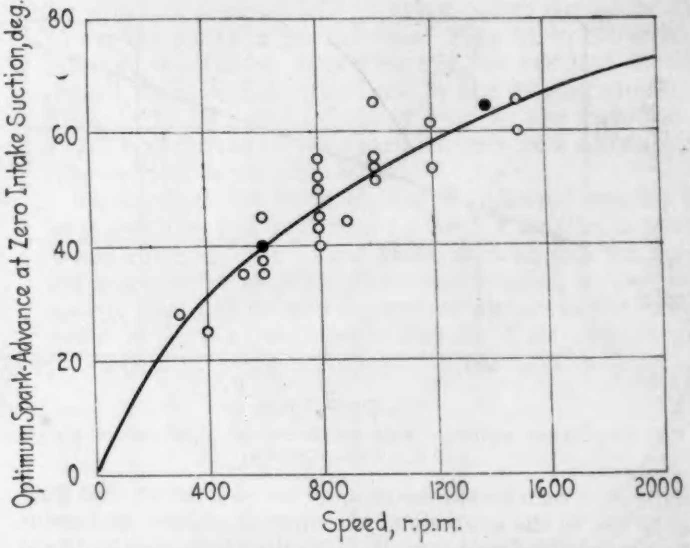

FIG. 9—EFFECT OF VARYING THE SPEED ON THE OPTIMUM SPARK-ADVANCE OF A FORD ENGINE

TABLE 3—OPTIMUM SPARK-ADVANCES ON FORD ENGINE

Speed R. P. M.	Intake Suction, In. of Mercury	Optimum Spark-Advance, Deg.	Tolerance in Spark-Advance for 10 Per Cent Power Loss
1,400	5.55	60	46 to 86
1,400	8.80	64	48 to 64
1,400	12.00	70	51 to 91
1,400	14.20	85	72 to 100
1,200	4.40	55	43 to 81
1,200	13.30	63	40 to 82
900	10.40	52	36 to 68
800	2.60	53	32 to 72
600	1.10	42	23 to 63
600	6.70	42	23 to 61
600	11.70	58	38 to 75
600	16.60	49	37 to 59

Some tests made on this engine, as a preliminary examination, have been presented already in Figs. 3 and 4. The determination of the optimum spark-advance was for some time a part of the students' work in the laboratory. Each day a series of combinations of load or intake suctions and speeds were investigated; at each load-speed setting a curve of the brake-torque versus the spark-advance with at least five points was found. Table 3 shows such a set of determinations in 1 day's work. These were obtained with the mixture always at or near the maximum-power value, the intake air and the discharge-water temperatures were 140 deg. fahr. and the barometric pressure was 29.35 in. After a considerable amount of data such as those in Table 3 had been collected, curves were plotted, as in Fig. 8, of the optimum spark-advance versus the intake suction for a constant speed. Such curves were used for reducing all the re-

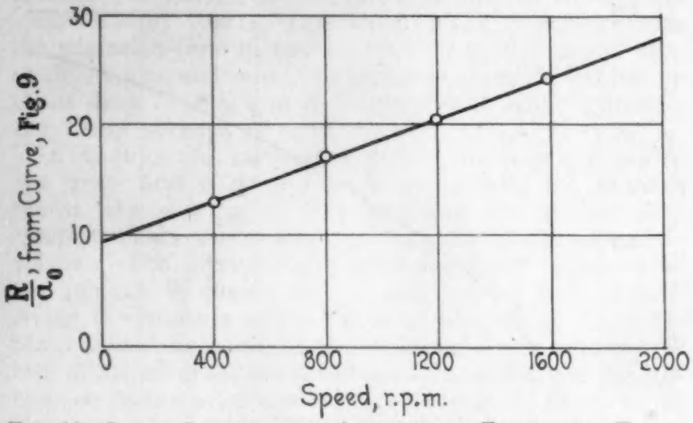

FIG. 10—CHART SHOWING THE APPROXIMATE TURBULENCE FACTOR FOR A FORD ENGINE

sults to a zero intake-suction, as we had not at that time gone far in the analysis of the dilution effect. Reduction to a zero intake-suction is reduction to a constant, not zero, volumetric dilution with spent gas, and to a mass dilution that is not constant at all speeds, because the exhaust temperature rises with the speed. The curves of Figs. 8 and 9 will illustrate, however, a very good method of analysis for commercial purposes. In Fig. 8 the effect of intake suction is seen to be, with sufficient accuracy, the same for all speeds; in other words, a curve of intake-suction effect can be drawn at the bottom of Fig. 8, and a curve parallel to this will answer for each speed. With the aid of this curve for intake-suction effect all the data were reduced to values for a zero intake-suction, and with these reduced values the curve of the optimum spark-advance for zero intake-suction versus speed in Fig. 9 is plotted. The optimum spark-advance for any combination of speed and intake-suction can then be found by adding together the advance-for-speed effect from Fig. 9, and for intake-suction effect from Fig. 8. For example, at 800 r.p.m. a 12-in. suction requires 47 deg. for speed and 13 deg. for suction or a total of 60 deg.

It is obvious that if this possibility of representing the optimum spark-advance in a given engine as a sum of two parts, one for speed and one for intake suction, is confirmed on other engines, we have a basis immediately for an automatic mechanical control of the spark-advance

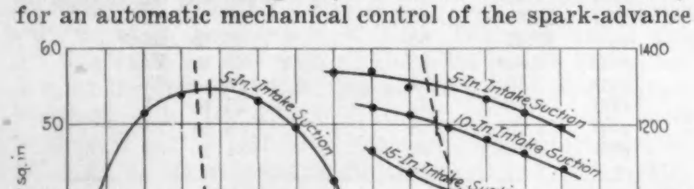

FIG. 11—STUDY OF THE DILUTION AND SPEED FACTORS IN THE SPARK-ADVANCE OF A CONTINENTAL ENGINE AT A SPEED OF 400 R.P.M.

that will keep it always very close to the optimum value. In the example just cited, the control for speed alone would have to be set for the full-load conditions, and if the automatic control by speed alone were not corrected by supplementary hand-control the error of spark-setting at the 12-in. suction would be sufficient to cause a loss of about 5 per cent in power and economy. As this loss occurs under the most common running conditions, it is worth thinking about.

Since we have not, in the curve of Fig. 9, entirely eliminated the effect of dilution, but have merely reduced it to a point where the dilution effect is nearly constant, we cannot, perhaps, determine the turbulence effect with complete accuracy. The form of the curve in Fig. 9 is due principally, however, to turbulence effect. Hence, in Fig. 10, we plot R/a_0 versus R, picking values of a_0 from Fig. 9 for various values

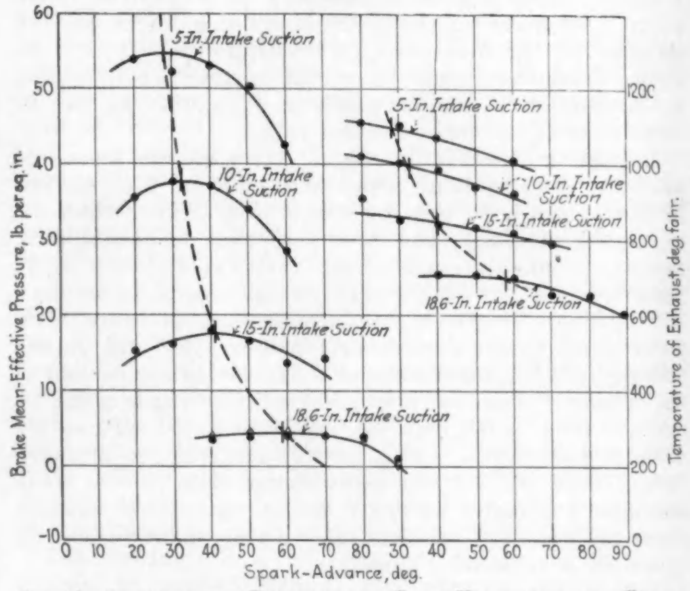

FIG. 12—STUDY OF THE DILUTION AND SPEED FACTORS IN THE SPARK-ADVANCE OF A CONTINENTAL ENGINE AT A SPEED OF 800 R.P.M.

of R. The curve in Fig. 10 is really only a graphic analysis of Fig. 9. The equation of the straight line in Fig. 10 is $R/a_0 = 9.3\ (1 + 0.0010R)$ Hence the equation of the curve in Fig. 9 is $a_0 = 0.108R/(1 + 0.0010R)$. Since $a_0 = (2/9)R$ times the explosion-time, it follows that the explosion-time of the Ford engine combustion-chamber as a bomb, without turbulence, but with the dilution corresponding to a zero intake-suction, is $(2/9) \times 0.108 = 0.0230$ sec.

It is rather gratifying to find, in Fig. 10, that turbulence in an engine corresponds to the same type of law of action as in the case of Hopkinson's bomb tests, as is shown by comparing Figs. 7 and 10; and that the turbulence factor for the speeding-up of the combustion by turbulence can be evaluated so readily by simple tests for determining the optimum spark-advance.

SUPPLEMENTARY TESTS ON A CONTINENTAL ENGINE

As Fig. 8 shows, the data obtained from student work on the Ford engine were not good enough to evaluate the dilution factor in spark-advance in a scientific way. Hence, we turned to a Continental 7-R six-cylinder engine, which we have set up in the Cornell University laboratory, with an electric dynamometer and a fairly complete outfit of measuring devices. A Sylphon thermostat maintains the discharge water at from 135 to 140 deg. fahr.; the carbureter is of Stromberg make, model M1; the intake air is at room temperature, and a large

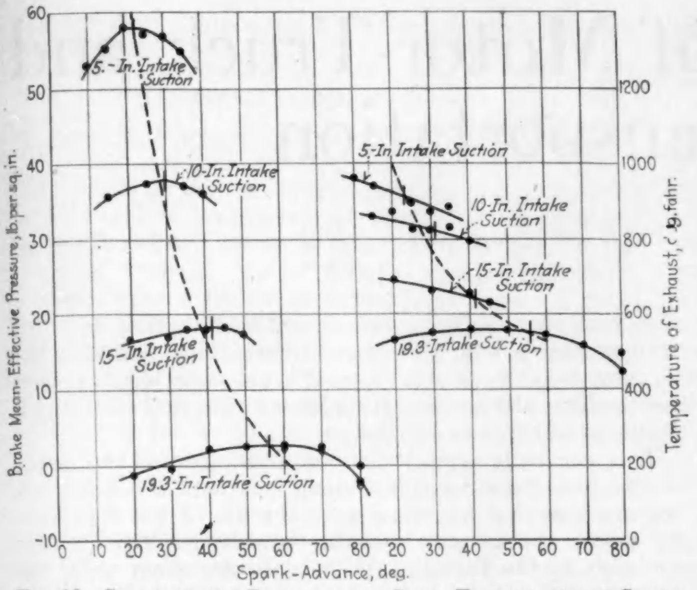

FIG. 13—STUDY OF THE DILUTION AND SPEED FACTORS IN THE SPARK-ADVANCE OF A CONTINENTAL ENGINE AT A SPEED OF 1200 R.P.M.

hot-spot incorporated into the manifold. The compression ratio is 4.55 to 1. At full throttle, this engine detonates considerably on Standard Oil gasoline and this fact was bothersome at times, yet led to one of our most interesting discoveries concerning the reaction-rate of combustion, or the effect of an anti-knock.

The main body of our experimental work on the Continental engine is shown in Figs. 11, 12, 13, and 14. At each of the speeds, 400, 800, 1200, and 1600 r.p.m., we

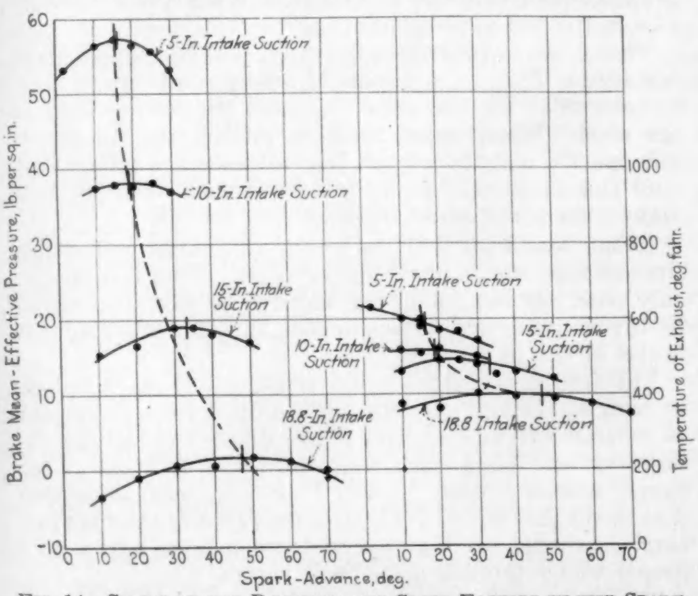

FIG. 14—STUDY OF THE DILUTION AND SPEED FACTORS IN THE SPARK-ADVANCE OF A CONTINENTAL ENGINE AT A SPEED OF 1600 R.P.M.

made sets of runs with the intake-suction at 5, 10, 15 and approximately 20 in. of mercury. In these runs we found the brake mean-effective pressure and the exhaust temperature as functions of the spark-advance. The optimum spark-advance can then be read off at the peak point of the brake-torque—mean-effective-pressure curve found for each combination of speed and suction resulting from a change of the spark-advance. Fairing curves represented by the dashed lines of Figs. 11 to 14 of the spark-advance versus the mean-effective pressure were then drawn in for each constant speed. The general results of varying the optimum spark-advance with the intake-suction or load are grouped in Fig. 15 and plotted for constant speeds. The observed points plotted in Fig. 15 are the peaks of the curves of Figs. 11 to 14, without fairing; the curves drawn in Fig. 15 are laid in with regard both to each other and to the fairing curves in Figs. 11 to 14. The curves of Fig. 15 are replotted in Fig. 16 as curves of the spark-advance at a constant intake-suction versus the speed.

Throughout the tests on the Continental engine the mixture-ratio was kept near to that of maximum power. When running near to the 20-in. suction any setting of the mixture for steady satisfactory running at the lower speeds was difficult and sometimes almost impossible; it could be done at the higher speeds. This circumstance

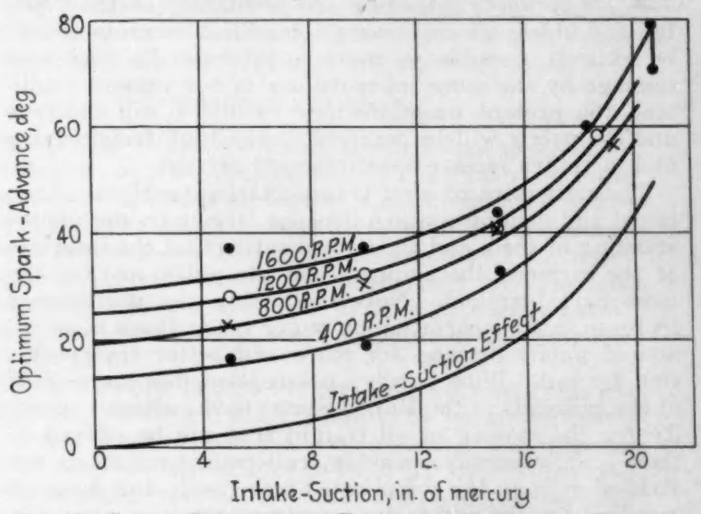

FIG. 15—CURVES SHOWING THE RELATION BETWEEN THE OPTIMUM-SPARK-ADVANCE AND THE INTAKE SUCTION FOR VARIOUS SPEEDS OF A SIX-CYLINDER CONTINENTAL ENGINE

probably is part of the reason that the spark-advance at the 20-in. suction must be so great at low speeds. The dilution of the charge with exhaust gas is, of course, very high at a 20-in. intake-suction; and it is higher at low speeds than at high ones, because the exhaust temperatures are higher at high speeds.

The exhaust temperatures found are plotted in Fig. 17 against the intake suction for constant speeds, and in Fig. 18 against the speed for constant intake-suctions.

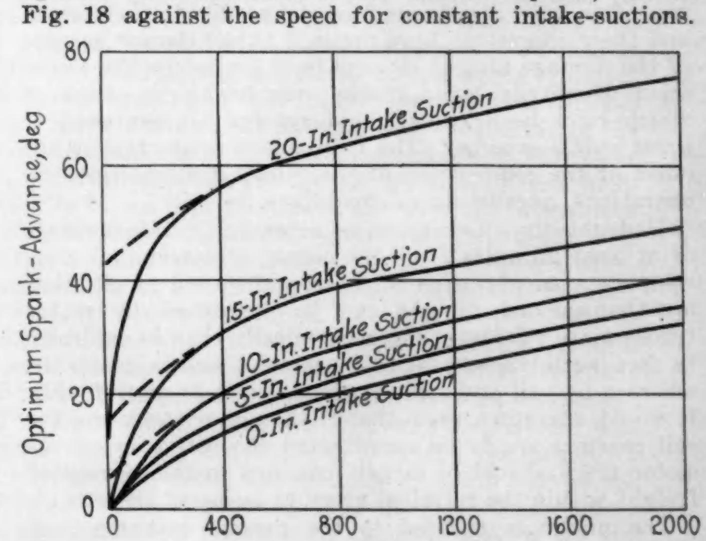

FIG. 16—CURVES SHOWING THE RELATION BETWEEN THE OPTIMUM SPARK-ADVANCE AND THE SPEED OF A CONTINENTAL ENGINE FOR VARIOUS INTAKE-SUCTIONS

(Concluded on p. 172)

The Coordination of Motor-Truck and Railroad Transportation

By W. J. L. BANHAM[1]

CLEVELAND TRANSPORTATION DINNER ADDRESS

TRANSPORTATION is the most important problem in the Country today. The lack of transportation is seriously impeding the production of the Country and unless we can create a condition whereby it will be entirely possible to move a substantially increased tonnage by the more intensive use of our present facilities, the present unsatisfactory condition will continue and the public will be required to pay high freight-rates and in return receive unsatisfactory service.

The settlement of most transportation problems of national and general concern depends largely on the understanding of them and on the realization that the interests of the carriers, the shippers and the public are for the most part identical. Notwithstanding the phenomenal increase in transportation by motor truck, there is an insistent public demand for more and better transportation by rail. With hardly an exception, the main lines of the railroads in the United States have sufficient capacity for the moving of all freight that can be offered to them and, generally speaking, rail transportation is restricted only by the complicated movements and delay of cars and freight within the terminal areas.

The congestion at the railroad terminals, while restricting the use of motor trucks, as supplements to railroad service within terminal areas, has opened up a field for their operation in competition with the railroads and has developed the use of motor trucks to furnish complete transportation between industries separated by what is deemed to be a reasonable trucking distance. The limit of a reasonable trucking distance is variously estimated. During the last 10 years, owners of trucks have undertaken to extend the limit to so-called long-distance trucking, often with disastrous financial results to themselves, and their operations have aroused public clamor because of the damage alleged to have been caused by the movement of heavily laden trucks over highways, many of which have been recently constructed or improved at great public expense. The railroads also are feeling the effect of the competition of these long-distance trucking operations, paralleling railroad lines.

Undoubtedly within terminal areas and within a radius of at least 25 miles from the center of a terminal area, complete transportation of merchandise and of all other less-than-carload freight can be furnished by motor trucks more efficiently and economically than by railroad. In fact, such transportation by motor truck is profitable, whereas by rail and cartage combined it is unprofitable. It would, therefore, seem that if the motor truck and the rail carriers are to be coordinated the activities of the motor truck should be largely confined to the moving of freight within the terminal areas or adjacent thereto.

The public is entitled to the prompt movement of freight, and the carriers, when performing this service, are entitled to just and reasonable rates. There are three forms of transportation, namely, railroads, waterways and highways. The public is entitled to that form of transportation which is most economical and should not be compelled to use any form of transportation that does not perform the service, from both a time and rate standpoint, to which it is entitled.

It is generally agreed that the highways and the motor trucks have been developed along most efficient lines, but the problem that has to do with the use of the highways by motor trucks still remains unsolved. This is due probably to the fact that there does not seem to be the proper relation between the highway and the rail carriers. This may be due to a lack of understanding as to the economic rights, in the interests of the public, of these two important forms of transportation.

The railroads of the Country are faced today with the most severe motor-truck competition covering short-haul freight movements, that tends to deplete further their much needed freight revenues. The motor truck is also hauling freight long distances, frequently at rates that are considerably higher than rail rates, and if inquiry be made as to why the motor truck handles long-haul freight the answer usually is: The motor truck gives the service, while the railroads do not.

This is not entirely true, but it is true in a great many instances. Rail carriers complain that motor trucks haul freight when the rates are high and the road conditions are good. When motor-truck conditions are not favorable, particularly in winter, the railroads are required to haul this class of freight, which further adds to their many transportation troubles.

It has been said that the public ultimately will define the economic range of the motor truck. This is undoubtedly true, but the public will have to be educated, either by the railroads or by the shippers, in order that they can arrive at the proper conclusion.

The highways and the motor truck can do much to help or hamper the rail carriers, but before improvement can be made it will be necessary to coordinate the rail and the highway and place each form of transportation in its proper economic field. It will, therefore, seem important that in the interest of the public the rail and the highway carriers should get together and endeavor to arrive at a proper understanding as to their rights.

The shipping public is being urged by the carriers to load cars to capacity and to unload and load promptly, which would mean the use of much additional equipment now idle, due to conditions over which the carriers do not seem to have control. There is a limit, however, as to the amount of freight that can be loaded on cars, which is controlled in a great many instances by commercial conditions. This is not true, however, so far as the prompt unloading and loading of cars is concerned. The carriers urge that this be done but make no recommendations as to how it can be accomplished. I shall therefore call to your attention the necessity for the closest cooperation between all parties in the interest of some method that would tend to speed up what we might call "terminal

[1] General traffic manager, Otis Elevator Co., New York City; and president, Associated Traffic Clubs of America.

transportation." This applies particularly to less-than-carload freight.

Terminal Transportation

The prevailing practice is to unload less-than-carload freight from cars into the freight-houses and to place carload freight on delivery tracks and then notify the consignee by mail of the arrival of the goods. Less-than-carload shipments remain in the freight-houses of the railroads an average time of 3 days; carload shipments remain on delivery tracks in excess of 2 days. A well-organized store-door delivery plan should make the freight-house space available for use not less than twice a day, instead of once every 3 days; thus the capacity of these facilities would be increased six-fold. The capacity of the delivery tracks should, under the store-door delivery plan, be increased four-fold.

The investment in freight-house and delivery-track space represents an item of expense that is frequently lost sight of by the railroads in their computation of the cost incident to the handling of the traffic involving the use of these facilities. Attention is, therefore, directed to the fact that this item alone, in some localities, represents an expense of $3 per ton on less-than-carload traffic and as much as $10 per day per car on carload traffic.

Obviously, therefore, an increase in the capacity of an existing facility results in a reduction in the overhead expense of operating the facility, provided that the increase in the capacity is utilized. With a reduction in the overhead expense and a substantial reduction in the labor cost of handling freight through freight-houses, the railroads could well afford to offer the consignee, as an inducement to avail himself of store-door delivery service, a rate that is substantially lower than he is now paying. Unless the plan is made attractive to the consignee the volume of traffic influenced by it will not be sufficient to afford the freight-houses and the delivery tracks any material relief. And by making available to the consignee a better service at a lower cost to him all grounds for criticism and objection would be removed.

If we are to speed-up the movement of the freight passing through the terminals it will be necessary for the carriers and the shippers to agree upon and develop a system of store-door delivery and collection service. This would mean the more intensive use of the present terminal facilities and would make it entirely possible to handle a heavier tonnage both inbound and outbound. It is important that the rail carriers have control of their freight terminals in order that they may clear them promptly of inbound and outbound freight, which does not seem to be possible under the present method of handling freight moving over the carriers' platforms and freight terminals.

What is needed at present is closer cooperation between the shippers and the receivers of freight and the rail carriers. The lack, however, of a definite plan for handling this class of freight has resulted in the freight's remaining in the carriers' terminals to the extent that there is more or less freight congestion at all times. When the congestion becomes acute, freight embargoes are issued by the carriers, that result in the partial stopping of freight movement until the terminals can be cleared. It further results in delaying the carriers' equipment, which is needed both at this and at all times.

The present cost of handling freight through the terminals warrants the carriers in taking all the steps possible toward the prompt removal of freight from their terminals. No storage charge is sufficient to reimburse the carriers for the freight held in their terminals, on account of the space occupied, the extra labor required through the consequent congestion and the cost of shipments lost or damaged by storage and congestion.

Can the motor truck be used to relieve freight congestion, thereby reducing the carriers' terminal expense, and to transport freight to and from the carriers' terminals to a greater extent than it is being used at present? Under a proper system and organization the motor truck should play an important part in speeding-up terminal transportation. To do this, however, it would be necessary for the carriers and the shippers to agree upon a store-door collection and delivery service, which seems to be the only method whereby freight can be moved to and from freight terminals in the shortest possible time. Store-door delivery would relieve freight congestion and delay, and should finally solve the carriers' terminal problem.

Practice in Canada

Canada has solved the terminal-congestion problem by the adoption of store-door delivery. Practically since the inception of railroads in eastern Canada carriers in the principal distributing centers have provided a cartage service to and from the freight terminals and the warehouses or the store-doors of the shippers and the consignees. Both services appear to be economically sound, even when viewed from the standpoint of the carriers' interest.

It is interesting to note the reasonable teaming charges in Canada, which are made possible by organized teaming under a store-door delivery system. The average cartage-rate on carload lots is $0.04 per 100 lb. and the average cartage-rate on less-than-carload lots is less than $0.06 per 100 lb., which covers delivery to the wholesale and the manufacturing districts, a radius of approximately 2½ miles from the terminals. The same cartage-rates prevail on outbound shipments moving from the shippers' warehouses to the carriers' terminals.

As already stated, the service in eastern Canada is governed by tariffs issued by the carriers and applies to practically all traffic coming under the generally accepted terms of merchandise and package freight. It contains many advantages to the carriers, the shippers and the consignees. With the exception of such consignees as have elected to do their own teaming and have so notified the carriers, no advice of arrival is given. The freight is billed to the teaming companies with charges to collect and is delivered from the terminal, or car if a carload, within a few hours of the unloading or the placing of the car on the team track. The teaming companies have their own superintendent in the freight terminal, whose business it is to see that the trucks are promptly dispatched with loads for delivery within certain areas or zones, with the result that freight usually is in the consignee's warehouse before an arrival notice could reach him by mail. All freight is moved through the carriers' terminals or from the cars with a minimum of delay and freight charges, and the receipt for the delivery of the freight is obtained from the consignees by the teaming companies.

The same principle governs outbound shipments. The shipper telephones the teaming company operating for the carrier over whose line he desires to ship that he has a load or half a load as the case may be. These orders are consolidated by the teaming company serving the district in question, the freight is picked up at the shipper's warehouse and the bill of lading is signed by the teaming company as agent for the carrier. It is then in transit, the carrier's responsibility under the system beginning at the shipper's warehouse door on outbound traffic and ending at the consignee's warehouse or store door on in-

bound traffic. No orders for trucks to be filled that day are accepted by the teaming companies after 3 p.m., as all outbound loads either must be in the carrier's terminals or in line for unloading before 5 p.m. The system contemplates loading all the cars and forwarding all the outbound shipments unloaded into the terminals each day.

It is estimated that approximately 95 per cent of the shippers and consignees in towns and cities where there is store-door delivery use the carrier's teaming service, and it goes without further enlargement that this service must be a fairly satisfactory one to the public. The fact that the teaming companies handle such a large percentage of the traffic enables them to arrange for the loading and to utilize their equipment to the greatest possible extent.

The use of the motor truck and its proper relation to the short haul is an important problem in which the public is vitally interested, but the proper solution has not been agreed upon up to the present time.

Field of the Motor Truck

I believe that the motor truck and the highways should be used for the handling of short-haul freight over distances up to 25 miles or distances to be agreed upon by both parties. This would relieve the rail carriers of the responsibility of handling short-haul freight, which is not only carried at a loss from the viewpoint of the rate, but also congests the terminals and restricts the use of the carrier's equipment that is much needed for long-haul traffic. The carriers and the motor trucks should agree to cooperate in the handling of freight over distances to be agreed upon from the territory in which it is uneconomical for the carriers to handle freight and where it is to the advantage of the motor truck to take care of the short haul.

There is no reason, so far as I have been able to ascertain, why the motor truck should not act as a feeder to the rail carriers, the truck operators acting as agents of the rail carriers and having their duties and responsibilities clearly defined by tariff rules and regulations. An arrangement, such as that outlined, within a short time would place the motor truck in a proper field and tend to eliminate the present unwise highway competition. Under this arrangement the public would be advised as to the rates to be charged; and I believe that a service would be rendered that would not only be satisfactory to the shipping public, but also prove financially satisfactory to both the motor-truck operator and the rail carriers.

The movement of less-than-carload freight between the carriers' sub-stations and main stations is also an important problem, inasmuch as it involves much expense on the part of the carrier; and in many cases the service rendered is unsatisfactory to the public on account of the delay experienced by the carriers' handling this class of freight in what is known as "transfer cars." It seems that the transfer of this class of freight properly belongs to the facilities offered by the motor truck; it would eliminate the use of the carriers' equipment for the moving of this class of freight and would benefit both the shippers and the carriers of the Country greatly.

Notwithstanding the fact that existing terminal facilities are admittedly inadequate to accommodate carload traffic, the railroads, at a number of terminal centers, are using cars for the interchange of less-than-carload freight. The use of cars for this service not only seriously delays the traffic, but adds to the congestion of the terminals. It is admitted that 3 days is the average time consumed in moving a car from the loading station of one railroad to the unloading station of another railroad at one large terminal center, while at another terminal center when the traffic is handled by automotive equipment, 95 per cent of the freight so handled is delivered to the forwarding line on the same day that it is unloaded from the cars by the receiving line.

In some cases railroads have built substations outside the congested district, where all less-than-carload freight is concentrated, unloaded and reloaded into cars for distribution to various points. It is believed that, instead of reloading this freight into cars, it should be handled by automotive equipment, even though that service should prove slightly more expensive than moving the cars, to the end that relief may be afforded to the terminals and the cars be released for more profitable service. Unquestionably, prohibiting the use of cars and terminals for this character of service would measurably increase the capacity of the existing facilities.

The remedy for this situation presents no serious obstacles. It is only necessary for the railroads to arrange for handling the traffic by truck or dray. This arrangement might well be left in the hands of the local representatives of the railroads and the responsibility for the prompt and economical moving of the traffic be placed on them. If it should be found that no local transfer-company is equipped to do the work, it would be time to negotiate with an organization that is rendering a satisfactory and economical service at some other point.

The transferring of less-than-carload inter-line freight is also a costly operation to the carriers and tends further to congest the carriers' yards and terminals on account of delayed equipment, and with the methods now used in loading and unloading inter-line freight it seems that the motor truck is especially adapted for moving this class of freight, by the use of demountable bodies and other automotive equipment. I am firmly of the opinion that the result would be a speeding-up of inter-line freight and that handling by automotive equipment would result in profitable operation for the motor-truck companies and give prompt service to the public instead of delayed service and at a decreased cost to the rail carriers.

Trap Cars

The movement of less-than-carload freight now being handled in trap cars is a problem that still remains unsolved, and I am satisfied that when full consideration has been given to the rights of the highways and the motor trucks, without interfering with the just rights of the rail carriers, the motor truck can and should be used for this special class of service.

What has been said with respect to the use of cars and terminal facilities for handling interchange less-than-carload freight between railroads will apply with equal force to "trap-car" service. The term "trap-car" is used to designate a service that is performed free-of-charge by the railroads for the shipper or the consignee with track location, on less-than-carload shipments weighing, or aggregating in weight, 8000 lb. or more. The practice is recognized by railroad officers as an unprofitable one and its discontinuance has been proposed by them on several occasions, not only because of its unprofitableness, but also because it gives to the shipper, or the consignee, located on a track, free store-door service on less-than-carload freight, an advantage not enjoyed by shippers or consignees without track location. Freight agents and terminal superintendents are almost unanimous in expressing opposition to the handling of less-than-carload freight in trap cars. Their opposition is based on the

(Concluded on p. 138)

Air-Cooled Automotive Engines

By C. P. Grimes[1]

METROPOLITAN SECTION PAPER — *Illustrated with* PHOTOGRAPHS AND DRAWINGS

THE author believes that the universal power unit will be direct air-cooled, but states that the direct air-cooled engine is now in the minority because, until very recently, there has not been a sufficiently broad series of established engineering facts and development work available to form a foundation for improvement. The satisfactory air-cooling of an 8 x 10-in. cylinder has been reported, and the development in a smaller cylinder of 138 lb. per sq. in. brake mean-effective pressure; also, in a three-cylinder, air-cooled, radial-engine, a brake mean-effective pressure of more than 125 lb. per sq. in. was developed and the engine endured beyond the ordinary expectations for water-cooled engines. Since these sizes and values are larger and greater than those of standard automobile engines, even though these developments may have been accomplished through the use of high-grade airplane-fuel, delicate aluminum finning and costly cylinder construction, the author feels justified in the conviction that direct air-cooling already is a success. The paper is devoted to a detailed and illustrated description and discussion of the engineering features of the car built by the company represented by the author.

THE direct air-cooled engine already has been developed to a state of perfection that should, in a very short time, allow it to prove the superiority of direct air-cooling. When constructed properly from recent designs it will be found that its exhaust-valves will be cooler, the carbon formation less and that it will be free from burnt and warped valves that so often are caused by the formation of a lime deposit in the water-jacket adjacent to the cylinder-head and the valve-seats. The universal power unit will be direct air-cooled because it already has been proved superior in the desert and for polar-expedition usage.

At present, the direct air-cooled engine is in the minority because, until very recently, a sufficiently broad series of established engineering facts and development work was not available to form a foundation from which the air-cooled engine successes of the immediate future could spring. S. D. Heron reports the satisfactory cooling of an 8 x 10-in. cylinder and the development in a smaller cylinder of 138 lb. per sq. in. brake mean-effective pressure, which is decidedly more than that developed by any standard automobile engine. C. Lawrance's three-cylinder radial-engine endured far beyond the expectation for any water-cooled engine and showed a brake mean-effective pressure of more than 125 lb. per sq. in. which, once again, is far higher than that used by any standard automobile engine. Although the above developments may have been accomplished through the use of airplane fuel, a delicate aluminum finning and costly cylinder construction, I feel confident that the results should warrant the conviction of the most skeptical that direct air-cooling is a success and must be recognized as such if further successful engine development is accomplished.

COOLING ABILITY

About 22 years ago, John W. Wilkinson applied his skill to the construction and operation of the first direct air-cooled car built by our company, and his work has resulted in our Series-10 car.

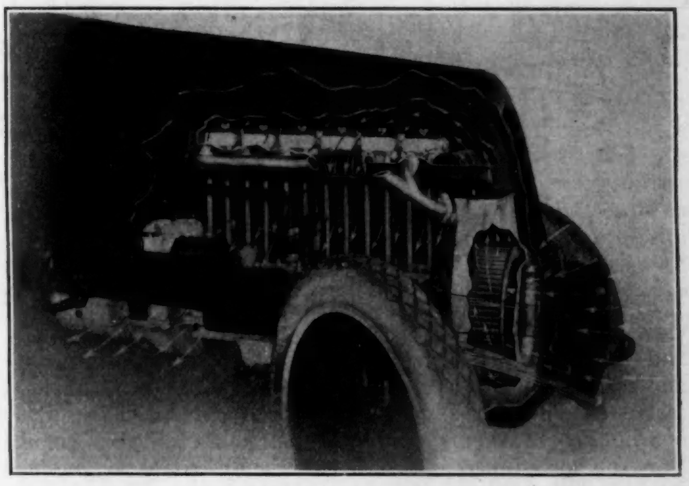

FIG. 1—PHANTOM VIEW OF THE COOLING SYSTEM USED IN THE FRANKLIN CAR

I acknowledge that the cooling system on some of the early models of our cars was not all that could be desired, but I believe the Series-10 cooling system to be at least the equal of any water-cooled system. Supposing a temperature of 100 deg. fahr. in the shade to be that of a summer day, that the powerplant of a Series-10 car, with all the gasoline tanks, the mufflers and the like, has been removed and mounted on a power jack to run a dynamometer or pump that will allow regulation of the load and that a stream of air or water is provided to play on the oil-pan; this engine can then be run with open throttle at a speed of 500 to 2500 r.p.m. as long as desired, without the necessity of using any additional air-cooling, I suggest that if the powerplant from a water-cooled car were tested in this same way, respect for the air-cooled engine would not be lessened. The Series-10 direct air-cooled powerplant has been operated from the Atlantic to the Pacific, far north and far south, through Death Valley and across the American Desert, and has demonstrated that the direct air-cooled engine has arrived.

Fig. 1 is a phantom view of the cooling system. The Series-9 engine used a Sirocco cooling fan 17 in. in diameter, of 3½-in. width of face and having 64 blades mounted on the flywheel to suck the cooling air down past the cooling fins, but the Series-10 fan is mounted at the front end of the crankshaft and blows the cooling air up through a duct over the top of the cylinder and then down past the cooling fins, exhausting into the hood of the car. The Series-10 Sirocco fan is 14 in. in diameter, has a 3½ in. width of face and 64 blades tipped 32 deg. forward the same as in the Series-9 engine. The blower system has been developed into a very efficient one that now delivers a greater volume of cooling air at a greater pressure, and yet consumes far less power to drive it, than does the Series-9 system, with the 17-in. fan. At 1600 r.p.m. of the engine, the 14-in. fan consumes 1 hp., as compared with 2 hp. for the 17-in. fan.

[1] M.S.A.E.—Research engineer, H. H. Franklin Mfg. Co., Syracuse, N. Y.

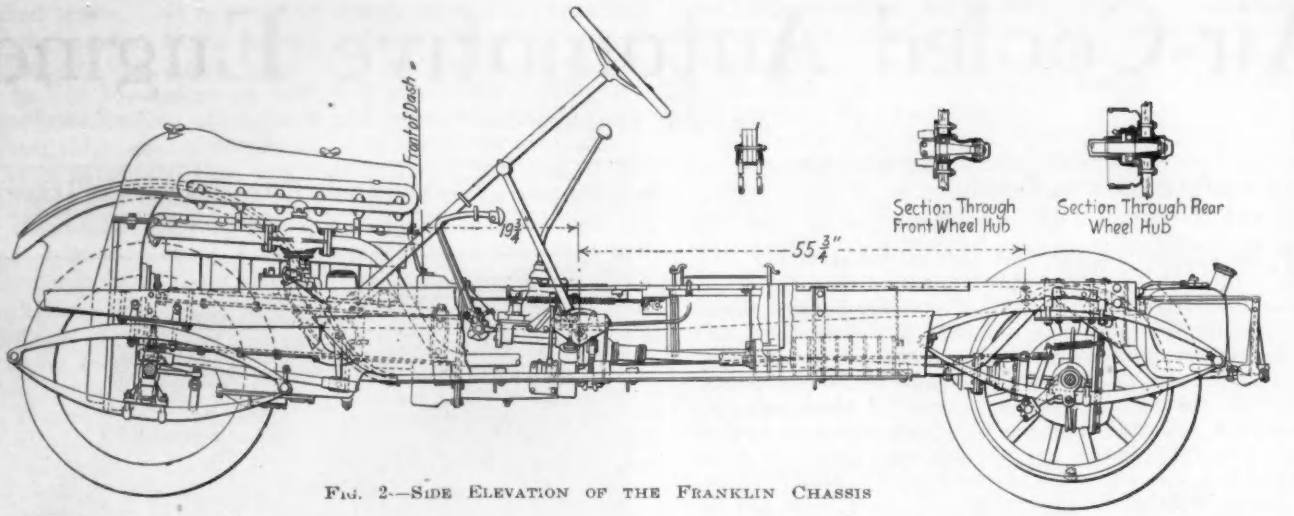

FIG. 2—SIDE ELEVATION OF THE FRANKLIN CHASSIS

Figs. 2 and 3 are, respectively, side elevation and plan views of the chassis. Since the chassis of the Series-9 and the Series-10 cars are identical insofar as weight and rolling resistance are concerned, I shall continue to discuss the cooling system in terms of car speed in miles per hour because my curves are all made on this basis. For the assistance of those who may wish to reduce these figures to a practical application, I have computed the diameter of an orifice of equivalent area to that of the total air-passages that are formed around the six cylinders of our engine. This equivalent orifice is computed to have a velocity coefficient of 100 per cent. The Series 10-A orifice has an area of 0.306 sq. ft. The Series 9-B orifice has an area of 0.238 sq. ft. The cylinders on the 9-B engine were built on $5\frac{1}{2}$-in. centers; but the cylinders on the 10-A engine are built on 5-in. centers. At a car speed of 20 m.p.h., the 9-B engine used 8 cu. ft. per sec. of cooling air as compared with 15 cu. ft. per sec. for the 10-A engine. The back pressure on the 9-B engine was 0.3 in. of water; on the 10-A engine it was 0.6 in. of water. Our Series 10-A car will cool perfectly at 5 m.p.h. with a full-open throttle at a temperature of 95 deg. fahr. in the shade for as long as it may be desired to drive up hill at this speed.

I will endeavor to show later, in detail, how we obtained the increased efficiency in the cooling system. The volume of cooling air delivered increases directly with the speed of the engine. The pressure of the cooling air in the duct increases as the square of the speed, while the horsepower consumed by the fan increases as the cube of the speed. We have no desire to use and do not believe in the ultra high-speed engine, but I feel it to be evident to all that it would not be advisable to attempt to cool with the direct-connected fan, since the power it consumes increases as the cube of the engine speed. The power consumed by our cooling fan at 20 m.p.h. is approximately 0.3 hp. At 30 m.p.h. it is 0.8 hp. After making careful comparisons of the power consumed by airplane propeller-blade fans just back of a radiator, I am led to believe that our cooling fan uses no more and, in most cases, decidedly less horsepower to drive it than is consumed by the ordinary fan just back of a radiator. Our grille construction provides a freer passage of air than would be possible with a water-filled radiator. All of the air pumped by our fan acquires a rise in temperature of from 90 to 120 deg. fahr. while passing over the engine, and may reach 220 deg. fahr. at the exit with perfect safety. Since the water-filled radiator never heats the air to more than from 45 to 60 deg. fahr., at least twice the volume would be required for a given cooling with a proportionate increased consumption of fan horsepower.

The water-circulating pump with its driving gear, shaft, bearings and stuffing boxes, can easily absorb as

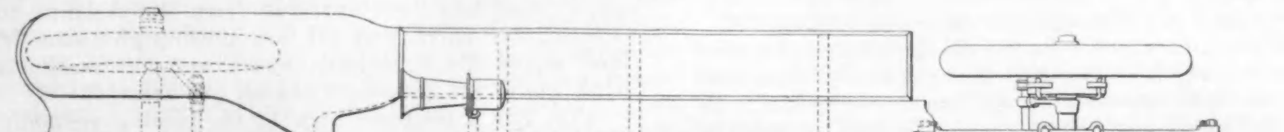

FIG. 3—PLAN VIEW OF THE CHASSIS SHOWN IN THE ELEVATION IN FIG. 2

much power as the water-cooling fan. The increased power losses and the additional complications necessary to operate a water-cooled system have been accepted by a majority of engine builders, but I am convinced that they must recognize the increased efficiency and the direct simplicity that we are able to gain through the use of a single fan wheel mounted on the engine crankshaft, as illustrated in Fig. 4.

When operating on the dynamometer, I have observed that our engine uses approximately 2 cu. ft. per hp. per sec. of cooling air at the lower speeds around 400 to 600 r.p.m. At the higher speeds, the engine uses from 0.9 to 1 cu. ft. per hp. per sec. of air.

Since, for many practical reasons, the fan must be located on the crankshaft, the problem at once presents itself how to obtain a sufficient pressure of air at extremely low engine speeds to cool the engine properly. Since it was not desirable to increase the diameter of the fan on account of general bulk and road clearance, we finally tipped the fan blades forward. This arrangement provided an air pressure equal to 1.68 times the peripheral speed of the fan. In other words, we are able to increase the air pressure for a given peripheral speed to the extent of 68 per cent over that which would be obtained by the use of a straight blade. Fig. 5 is a side view of the engine assembly. Fans with different numbers of blades were tried of course.

TABLE 1—COMPARISON OF COOLING-SYSTEM WEIGHTS, SIX-CYLINDER CARS

Name of Part	Our Car, Lb.	Other Cars, Lb.	
		Car *A*	Car *B*
Fan	14.47	3.2	2.0
Upper Air-Hood	18.31	......	......
Lower Air-Hood	10.66	......	......
Front Fan-Housing	8.25	......	......
One-Half of Rear Fan-Housing	6.56	......	......
Cylinder-Block and Valves	92.64	118.0	99.0
Cylinder-Head	......	......	42.0
Radiator Core and Shell	......	49.0	54.0
Water-Pump	......	7.5	10.0
Water Hose and Pipes	......	2.5	1.0(*a*)
Fan Support and Belt	......	2.0	0.7(*a*)
Thermostat	......	......	2.0
Water in System	......	31.4	37.6
Fan Driving-Pulley	......	2.0(*a*)	2.0(*a*)
Total Weight	150.89	215.6	250.3
Cylinder Size			
Bore, in.	3¼	3⅜	3⅜
Stroke, in.	4	4½	5
Displacement, cu. in.	199	242	268
Weight in Pounds Divided by Cubic-Inch Capacity of Cooling System	0.754	0.891	0.933

(*a*) Indicates approximate weights.

The next problem was to flow the air into the fan so that it would pass through in an efficient manner. The ordinary sharp-edge square-cornered entrance was found to be impossible because the air did not fill the full width of the blade. Our experiments showed one-third of the blade width to be inactive. We overcame this by using a scrolled-edge entrance, formed with as large a radius as was practicable. A very large radius would increase the entrance losses. A very small radius would increase the fan losses.

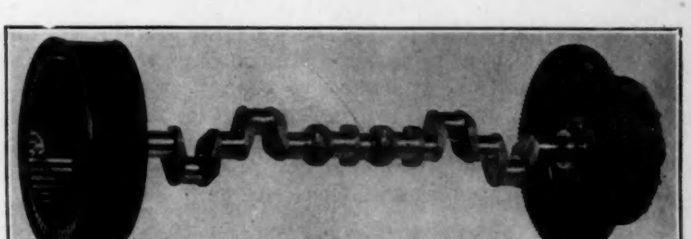

Fig. 4—A Single Fan Wheel Mounted on the Engine Crankshaft Supplies the Cooling Air

Experiments were next conducted with various scrolled shapes in an endeavor to have each square inch of blade area do an equal amount of work. The scroll housing and aluminum ducts were next developed by the elimination of eddies. The distribution of the cooling air so that an equal volume would pass by every cooling fin on each cylinder was accomplished through the use of deflectors in conjunction with the proper shape of the duct. It once required 14 separate pieces to obtain this distribution, but we were able to reduce this number to three pieces of very special shape and yet obtain a uniform distribution of the cooling air with a minimum loss of pressure. The present cooling system draws the air directly through the lower part of the grille at the front of the car, and passes it out through the cylinder fins into the automobile hood. At this point the heated cooling air is diluted by a large volume of free air that is allowed to enter the upper part of the car hood to dilute the hot air and reducing its temperature to a point slightly above that of the surrounding atmosphere, after which it all passes out underneath the car.

It must be evident to all that a direct air-cooling system that would cool properly at 110 deg. fahr. in the shade must over-cool at 10 deg. fahr. below zero. With this extreme in mind, I developed a thermostatic control of the cooling air. This control was operated through the use of a bi-metallic strip, screwed into the cylinder-head, which controlled a connection between the suction-yoke and the operating bellows for the dampers. This system gave remarkably close regulation on a car to the extent that the cylinder-head temperature could be maintained at any predetermined point within the range of ± 25 deg. fahr. This system also was arranged so that, in case the orifice became plugged-up or a pipe became broken, the dampers would be opened by the force of a spring and would remain in this position until the difficulty could be overcome. This idea was of great as-

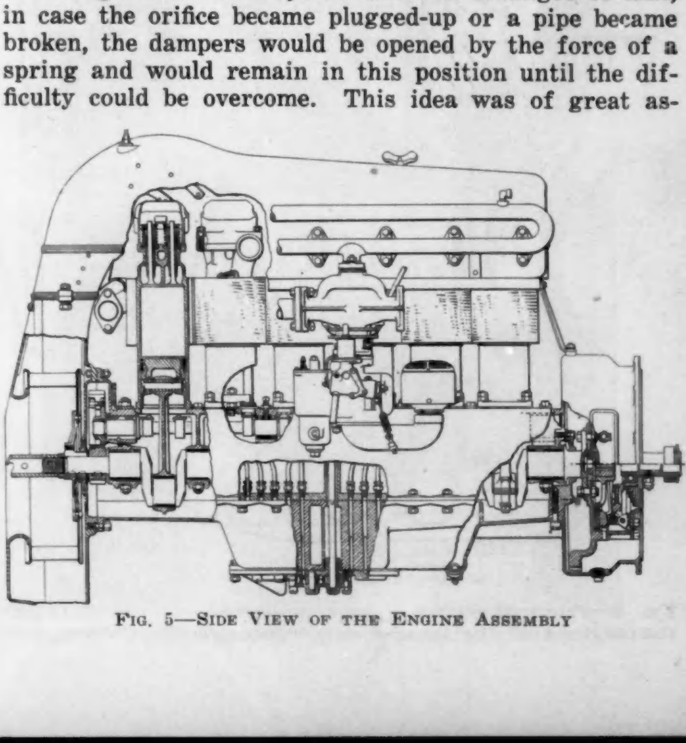

Fig. 5—Side View of the Engine Assembly

sistance on a cold morning in warming up the engine. It showed also a remarkable improvement in gasoline economy. In fact, it looked to be a very efficient and necessary device until we developed the exhaust-heated vaporizer, that maintained the temperature of the suction-yoke gases at 160 deg. fahr. ± 5 deg. throughout the entire working range of the engine.

Comparison of Cooling System Weights

A considerable amount of verbal comment has been made, but I have not observed any written comparisons regarding the relative weights of the cooling systems of air versus water. It is difficult to get a truly equitable basis. In this case I selected two prominent six-cylinder engines and have included the complete weight of their cylinder-block and valves against the complete weight of our cylinders with fins and valves, as shown in Table 1.

Since our total weights as well as the weights per cubic inch of displacement are both decidedly less, I am convinced that the direct method of cooling shows a decided advantage in this respect.

Details of Construction

Next to the cooling system, I feel that the cylinder is the most important part of any direct air-cooled engine; it is shown in Fig. 6. The cylinder is designed on 5-in. centers, has a 3¼-in. bore and a 4-in. stroke.

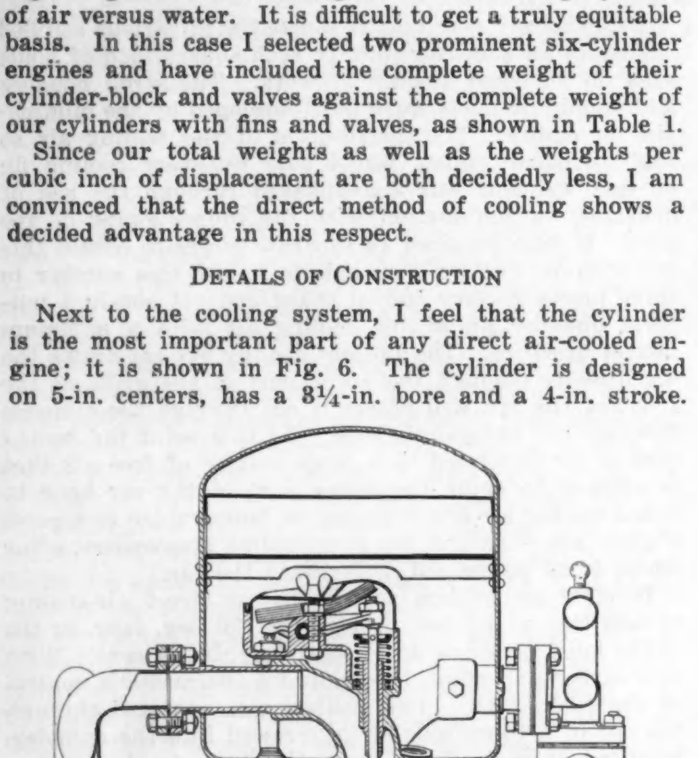

Fig. 6—Front Elevation Partly in Section of the Franklin Engine Showing the Arrangement of the Cylinder Construction

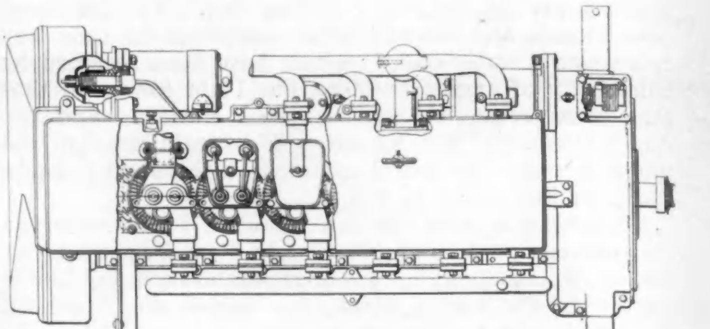

Fig. 7—Plan View of the Engine Assembly

Each cylinder with its valve and springs weighs 15.4 lb. The cylinder fins are 52 in number; they are 5 in. long and 1⅛ in. wide; 1/16 in. of this width is cast into the cylinder and 5/16 in. of it is turned over on the outer edge so that each fin can form its own jacket. This arrangement allows the jacket to assume the role of direct radiation. Each cylinder with its fins and exposed heat area has a radiating surface of 650 sq. in. While passing through the development stage, we built cylinders of aluminum, of bronze and of various combinations of metal. The cylinder shown in Fig. 6 has proved itself to be preferable in every respect to all the others that we have tried. When using a cylinder fin of greater length, we observed an increase of the weight and an increase of the air resistance, but did not observe a proportionate increase in the cooling ability because it is just possible that the air reached a very high temperature before it had traversed the complete length of the fin. We tried fins of one-half the present thickness, which would be 0.026 in., but found them to be impracticable because the air passages were so small that mud, grasshoppers and bugs closed the opening and made the engine overheat exactly in the same way as it would do with any water-cooled radiator.

Foundry Production Practice

As to foundry production methods, a steel cylinder with machined grooves along its length is first loaded with fins that are held in place by an ordinary soft-iron wire. This is placed in the mold and sand is allowed to fill-in between the fins. The steel cylinder is then very carefully extracted, exposing 1/16 in. of the edge on all of the fins. The suction and the exhaust ports and ducts are formed in cores and added as a unit. The cylinder is poured on-end with the head down. No heat-treatment is used in the process of the construction of this cylinder. When completed, the combustion-chamber has been machined and the bore ground to size; there is a wall thickness of 3/16 in. under the steel fins. This thickness is very necessary to give assurance that no hard spots appear along the bore. Before delivering the cooling fins to the foundry we found it advisable to tin them all over, which then assures us of practically a 100-per cent union between the fins and the cylinder-walls. The head of our cylinder is ⅝ in. thick. We have tried heads decidedly thinner and some 2 in. thick. We met with absolute success on all heads thicker than ⅝ in., but found that the extremely thin heads did not provide a sufficiently uniform temperature around the exhaust-valve seat.

Those who have endeavored to produce a quiet overhead-valve mechanism will appreciate our problem when I say that the expansion of our cylinder alone amounts

to 0.019 in. and that the exhaust-valve stem adds another 0.007 in., making a total of 0.026 in. that must be compensated for to maintain a uniform rocker-arm clearance at the valve-stem. The assembly shown in Fig. 7 provides for a normal adjustment of 0.010 in. at room temperature and a maximum variation of $\pm$ 0.002 in. within the range of cylinder temperatures from zero to 400 deg. fahr.

We mount our rocker-arm housing on top of the cylinder just beyond the valve-stem on one side, and we support the opposite side on two metal tubes that are concentric with the push-rod. This forms a semi-flexible arrangement, that can go-and-come with the expansion of the parts, yet it does not change the tappet clearance. The valve-springs on our engine "shimmy" at a number of different speeds. I have tried four different sizes of wire and two different diameters of spring, also one taper spring, but cannot seem to get rid of this peculiar action. I make a special appeal for suggestions as to

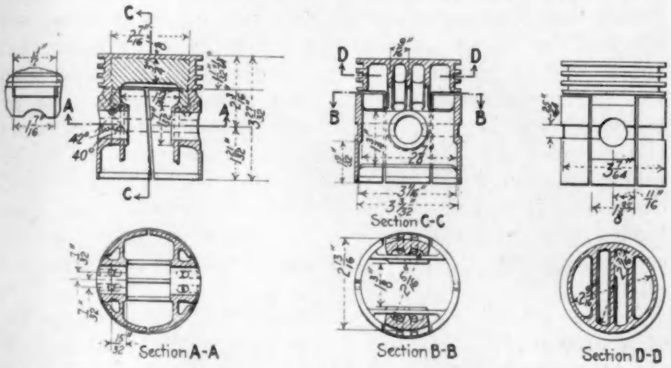

FIG. 8—ELEVATION AND CROSS-SECTIONS OF THE PISTON

how to overcome the shimmy or period points best in a plain valve-spring.

Fig. 8 shows section views of the piston. We have developed three distinct types of piston construction suitable for production. In all three an attempt has been made to isolate the ring-carrying portion from the skirt portion. Two of the types are sawed through and one is cast exceedingly thin at one point. The two that are sawed through have heavy supporting ribs under a 1/8-in. head; the third has an exceedingly thick head-construction. At present, I feel that the thick head-construction is less apt to collect carbon on the top, and shows itself to be almost entirely free of oil burning underneath. The true cylindrical non-split type seems to run with the least friction. The question of being able to free it permanently from sticking or scoring on the one hand, and from slaps on the other, can be solved only by trial. At present, and for a number of months previous, we have had remarkable success with a split-skirt type in which we use three rings. The top one is a plain concentric snap-ring; the middle ring has a special three-piece construction; and the bottom ring is a one-piece concentric snap-ring machined with a sharp edge for wiping oil. Piston and ring theories are numerous. We want the rings to wear-in quickly and to maintain a perfect bearing, and yet not to wear-out or break-out. Some types are far superior to others, but I feel that the technique of fabrication is of more importance than the design alone.

The forged duralumin connecting-rods shown in Fig. 9 are the result of a long series of experiments on the shape and the area of the various sections. Our steel connecting-rod would weigh 1 lb. 15 oz. complete, but our duralumin forged connecting-rod now weighs only

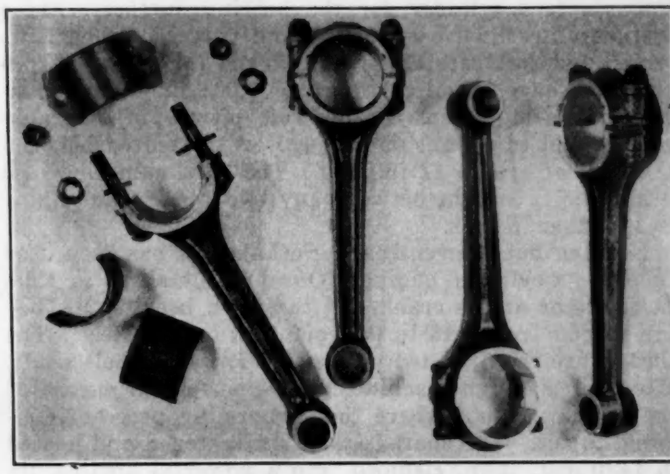
FIG. 9—VIEW SHOWING THE PARTS ENTERING INTO THE FORGED DURALUMIN CONNECTING-ROD AND THE ROD COMPLETELY ASSEMBLED

1 lb. 4 oz. complete. The piston-pin floats in the rod and in the piston. The hole in the rod is reamed to within 0.0005 in. of the finished-diameter in two operations. The hole is then burnished to size the same as is done in the piston. This produces a very superior finish in the hole, that seems to possess remarkable wearing qualities. After the die-cast bushings are locked by grooves into the big end, a combined swaging and burnishing operation packs the babbitt and produces an intimate contact with the rod at all points. The cap is assembled with babbitt-faced shims, after which a fly cutter pro-

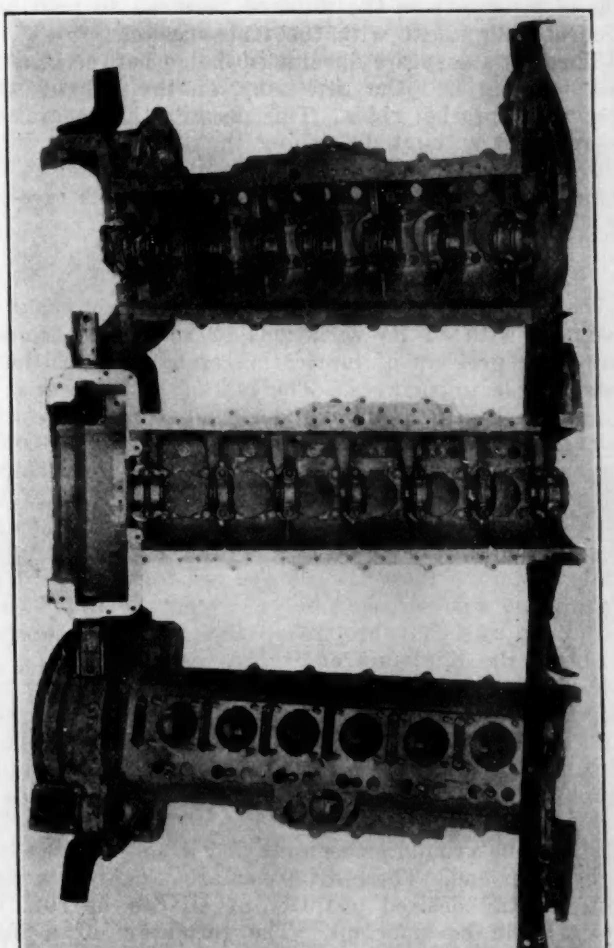
FIG. 10—THREE VIEWS OF THE CAST ALUMINUM CRANKCASE

duces a true cylinder for the bearing surface. These cylinders are in five diameters and vary by 0.00025 in., which makes it possible to select a rod for each crankpin that will have an oil clearance of 0.001 in. When completed, the alignment of this rod, piston-pin to crankpin, is held to a limit of $\pm$ 0.005 in., measured between the ends of arbors 12 in. long. The maximum allowable difference in weight between any two rods in an engine is less than 0.2 oz.

Nothing but a seven-bearing crankshaft has been used in our six-cylinder engine. Our present shaft is 2 in. in diameter at the crank and the main bearings. They are forged from 0.15 to 0.20-per cent carbon-steel. The throws are then twisted to position and the whole shaft, after being rough machined, is copper-plated except at the bearing points where the carburizing process is allowed to act. The shaft is then straightened and heated to 1420 deg. fahr., clamped into a powerful holding jig and quenched in water. It is then finish-ground to size. The case formed at the bearing points is 1/16 in. deep and registers 75 to 85 on the scleroscope. This case-hardened shaft, working against soft babbitts, will not need bearing attention until after 50,000 miles of running.

The crankcase shown in Fig. 10 is cast of aluminum, is unusually deep and is split 2⅝ in. below the shaft center-line. The seven bearing-caps are die-cast and held in place by through bolts seated against pressed-steel plates located on the top between the cylinders. The seven main bearings and the four camshaft bearings are rigidly supported by deep-ribbed sections. The main bearings are fly cut and reamed to size. The bell housing is fly cut from the finished bearings to provide an accurate alignment with the transmission.

There is a separate duralumin flange between the cam gear-housing and the generator, on the hub of which the chain sprocket rides. This permits the generator to be removed without disturbing the chain drive; it also allows a quick and accurate chain adjustment. The chain tension can be measured through a hole especially provided for this purpose.

LUBRICATION

When operating an air-cooled engine throughout the Country, with all its variations of roads and temperatures, the problem of lubrication assumes a position of considerable importance. The lubrication system is indicated in Fig. 5. Our cylinder-head temperature often reaches 400 deg. fahr. Our experience in the air-cooled-engine business has convinced us that the lubricating oil for our engine should be as free as possible from carbon. For this reason we are continually inspecting samples of oils from all of our dealers as well as from the principal vendors. This has allowed us to specify to our dealers and customers an oil that we have found to be absolutely uniform throughout the United States and to obtain the minimum of carbon residue.

A simple oil test can be conducted as follows: Obtain two bulbs from an electric fire-place heater, they will be about 2¼ in. in diameter, 12 in. long and should consume 500 watts. Mount them on a slant with the socket-end up. Arrange the oil so that it will drip on to the upper end of the bulb. That portion not vaporized will continue around either side and drop from the bulb at the lower end. These drippings are caught on a clean blotter. Any desired quantity of oil can be run, but 8 oz. should be sufficient. The formation of a black sticky compound on the blotter can be taken as an indication of the carbon residue from that particular oil that would probably lodge in the combustion-chamber as a carbon deposit.

Our engine has a seven-bearing crankshaft and is supplied with oil by a force-feed distributor-system. The oil-pump is driven from the camshaft and is located in the base of the oil-pan. Oil is drawn upward through a bronze screen that has an area of approximately 50 sq. in. From the pump distributor-plate, oil leads extend to all of the seven main bearings and to the chain gear bearing on the generator support. Each connecting-rod receives oil under pressure from its adjacent main bearing. The oil-pan is of cast aluminum and has corrugations in the bottom that are a material aid in maintaining the oil temperature at a reasonable figure of 160 deg. fahr. on a hot summer day. The one outstanding feature of this particular oiling system is the fact that, if for any reason one of the connecting-rods or one of the main bearings should burn out, the volume of oil that would be forced to the remaining bearings would not be altered. It has happened in our experiments that this particular feature was of great value and allowed the car to run all day with the knowledge that there was no chance for injuring anything, more than the particular bearing that had already gone. In an interesting experiment with baffle-plates located between the cylinder and the crankcase, we found that the baffle-plates increased the oil consumption at high speed and decreased it at low speed. We observed also that those baffles that had the smallest openings for allowing the connecting-rod to pass through showed the greatest effect.

For a given uniform extremely hot oil-temperature of approximately 205 deg., the pressure of the oil at both the pump and in the lead increases as the first power of the engine speed. At a uniform engine speed of 1800 r.p.m., the volume of oil forced through the bearings does not change over our practical or wide range of oil temperature and, at the same uniform engine speed, the oil pressure decreases with an increase of the oil temperature. In other words, our pump meters out a definite quantity of oil and forces this quantity to the bearing, regardless of the pressure required or the temperature of the oil pumped.

The automobile builder states how far his car will run on a gallon of oil; he says that if the oil-pan holds from 5 to 8 qt. the oil should be changed each 500 miles. When operating our car over 100 miles for each consecutive 3 hr., the oil consumption is 400 miles per gal.; when operating ordinarily, it is 800 miles per gal.

The oil-pump is arranged so that we can supply any desired quantity of oil to our engine. We have operated our cars successfully at the rate of 1200 miles per gal. but we feel that this is very inadvisable and have established the rates just given. I do not understand how an owner can profit by reducing his oil consumption to the minimum, in some cases practically starving the engine. We have been working on devices for the elimination of crankcase dilution for the past 2 years and have developed a unit that we believe will eliminate all crankcase dilution.

EXHAUST-HEATED VAPORIZER

While working for an improved fuel-economy, I developed an automatic vacuum-controlled spark-advance. Our ignition is now controlled by a mechanical governor that is very carefully calibrated for all conditions of full-throttle work. The vacuum-actuated bellows worked against a calibrated spring to increase the spark advance in the proportion of 1 divided by the absolute compres-

sion-pressure in the cylinder. This device improved the economy from 5 to 18 per cent. Next, I developed and installed a thermostatic control for the cooling air. The bi-metallic strip located in the cylinder-head allowed a connection with the suction yoke to operate a sylphon bellows connected in turn to dampers in the air duct. This operated very satisfactorily; I could control the cylinder-head temperature to ± 25 deg. fahr. When set for from 400 to 450 deg. fahr., the economy was increased materially.

My next development was an exhaust-heated vaporizer, shown in Fig. 11, in which a return trap for all wall-flow of fuel not vaporized on the first trip through was incorporated. The vaporizer performed better than my expectation, a glass tube located above and below showed clearly that all of the fuel was actually converted into a vapor. A thermocouple located at the yoke exit and

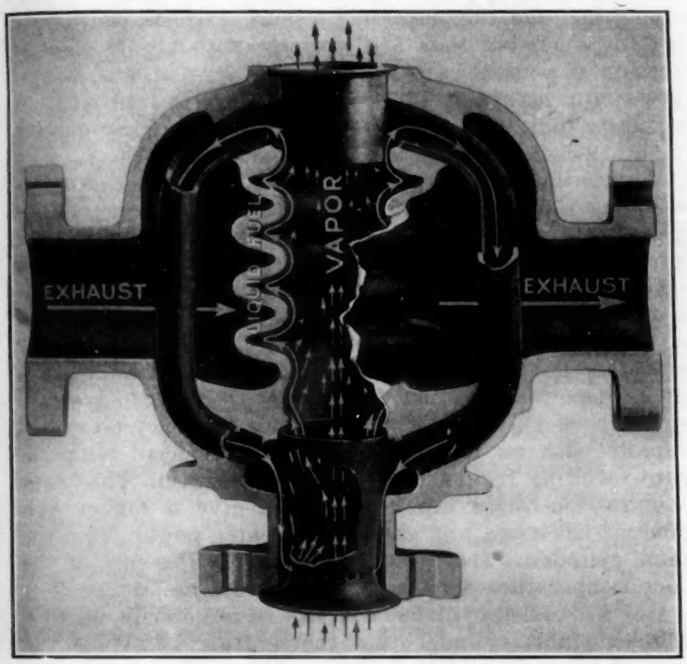

FIG. 11—AN EXHAUST-HEATED VAPORIZER IN WHICH A RETURN TRAP FOR ALL WALL-FLOW OF FUEL NOT VAPORIZED ON THE FIRST TRIP THROUGH IS INCORPORATED

suspended in the middle of the gas stream was used to determine the yoke temperature which remained at 160 deg. fahr. ± 5 deg. over all conditions of throttle and load at all speeds except those of less than 20 m.p.h., at which speed the yoke temperature was approximately 15 deg. fahr. cooler. Continued research with the vaporizer soon convinced me that, with it installed, I could no longer show an appreciable saving in economy either with the vacuum spark-control or the thermostatic temperature-control of the cylinder-head. This has proved out very well commercially, because we were able to install a plain casting having no moving parts that replaced two mechanical devices.

Our carbureter, shown in Fig. 12, is very simple. It consists of a plain gasoline nozzle controlled by an adjustable needle from the cowl. This nozzle is located in the primary choke. Extra air is supplied through a plain hinged valve for auxiliary air that has an adjustable spring. The float mechanism is arranged so that the weight of the float, by a lever, lifts the gasoline needle from the seat. We find the practical operation of a single large easily variable fuel-orifice offers many advantages; chief among them is our ability to adjust immediately

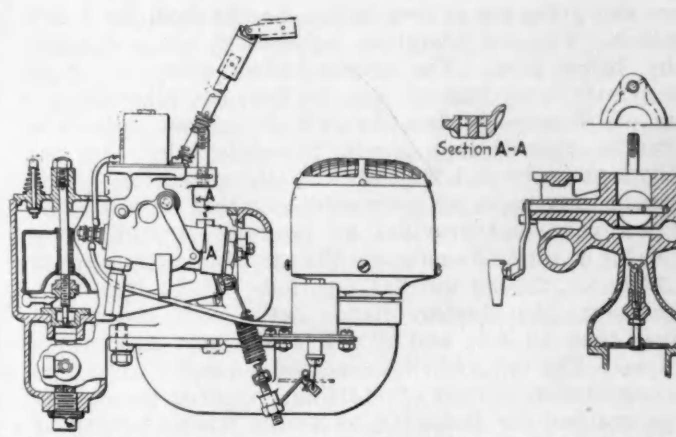

FIG. 12—ELEVATION PARTLY IN SECTION AND A CROSS-SECTION OF THE CARBURETER

for any change in the fuel quality and to be able to free the orifice from water or dirt readily when occasion demands.

Special efforts have been made in the design of the carbureter to facilitate starting in cold weather; to this end, for starting, the gasoline is vaporized and superheated electrically. This feature can be best explained by referring to the cut-away view in Fig. 13.

A miniature valveless carbureter has been constructed adjacent to the main instrument and draws its fuel from the common chamber. All the gasoline and the priming air pass over an electrically heated coil and are converted into a fixed gray vapor. This joins with the auxiliary air and passes up through a solenoid-controlled valve into the suction yoke just above the throttle disc that is always closed during this period.

To start in cold weather, the method is to close the throttle tight, open the gasoline needle-valve one entire

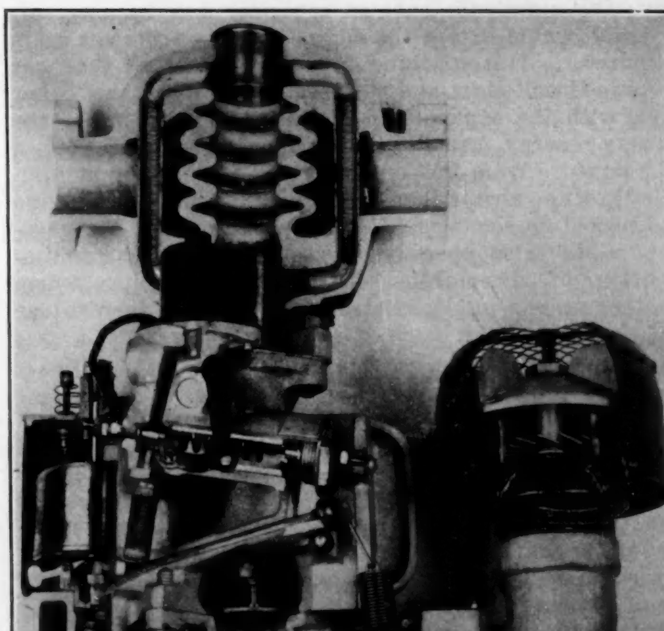

FIG. 13—A VIEW OF THE CARBURETER PARTIALLY CUT-AWAY TO SHOW THE ELECTRIC HEATING ARRANGEMENT PROVIDED FOR USE IN COLD WEATHER

turn and press the primer button on the dash for a few seconds. The last operation requires 40 sec. at 20 deg. fahr. below zero. The primer-button actuation opens the primer vapor-passage into the yoke and allows a current of 80 amp. to flow through the heater coil. The latter is enclosed in a special porcelain tube; one end of both the tube and the coil is touching gasoline. This forms a gray vapor or permanent gas that soon fills the suction yoke and provides an ignitable mixture in a sufficient quantity to maintain the car running at a speed of 18 m.p.h. for an indefinite period. At 20 deg. fahr. below zero, the starting motor never need be applied longer than 10 sec., and it will often start the engine in 5 sec. The old cold-choke method, as well as the partial-combustion system of starting, requires the engine to be cranked for from 1½ to 3 min. when starting at 20 deg. fahr. below zero. Our engine requires only from 5 to 10 sec. of cranking, after which the car can be run immediately at 18 m.p.h.

TABLE 2—COMPARISON OF STARTING CONDITIONS AT 20 DEG. FAHR. BELOW ZERO

	Our Car		Other Cars	
	Amp.	Sec.	Amp.	Sec.
Preparation for Starting.......	80	40	400	90
Actual Starting...............	400	10	400	10
Total, amp-sec.............	7,200		40,000	

A comparison of the starting conditions is made in Table 2. It shows a direct saving for our starting system of 32,800 amp.-sec. out of 40,000; or a saving of 32,800/40,000 = 82 per cent of battery energy. After the engine has fired for 1 min. at 20 deg. fahr. below zero, the aluminum exhaust-vaporizer has become sufficiently heated to run the engine directly from the main carbureter. Our machine is a dependable all-year car because it will start and run, even at 20 deg. fahr. below zero, with the expenditure of less than one-fifth of the battery energy usually required. The engine gets a rich mixture from the vaporizer and fires slowly at first. The throttle-valve is then opened slightly until the engine has picked-up somewhat. As soon as possible, the gasoline needle-valve opening is decreased to the running position of five-eighths of one turn. During warm weather and in mild climates, it is not necessary to use the electric primer.

Dust Separator

Since our carbureter has been arranged to pass all of the air through a common opening, it was easy to provide it with a dust separator. Its total weight is 14 oz. and it is illustrated in Fig. 13. This separator consists essentially of two units, both of which are mounted on the same shaft and rotate together. The lower unit is a windmill type of motor that receives its power from the air flowing to the carbureter. This causes the upper member or centrifugal pump to inhale into the housing a far greater volume of air than is used by the carbureter. Since the rotor disc with its radial vanes travels at high speed, all dust particles are thrown centrifugally against the side walls of the housing. At the lower edge of this housing is an opening about ⅛ in. wide extending clear around the instrument.

Since the centrifugal pump has inhaled 30 per cent more air than can be used by the engine and has forced all dirt particles by centrifugal force into this excess air, the dirt is carried away in this excess air, out through the bottom of the housing, and discharged to the atmosphere. As a test, I secured fine dust-particles from a dust separator; these particles had all been air-floated previous to their separation. I fed a definite quantity very slowly into the separator, which was operated at low speed with 1 in. of water suction. Careful weighing, in two tests, convinced me that this separator actually would clean from 96 to 97 per cent of the extremely fine dust particles from the airstream. This separator has greatly increased the life of the bearings and decreased the amount of carbon deposit in the combustion-chamber. The rotor runs fast enough to enable it to separate the dust at all car speeds above 3 m.p.h.

Operation

Our engine cylinder will not give satisfactory operation when the compression pressure exceeds 80 lb. per sq. in. gage, but they are working with absolute satisfaction at a speed of from 400 to 2500 r.p.m. and a compression pressure of 75 lb. per sq. in., measured with an accurate indicator. This refers to the use of low quality plain 18-cent gasoline.

I recently drove a car that showed rough engine operation and was informed that, whenever a compression pressure of 92 lb. per sq. in., or a brake mean-effective pressure of 97 lb. per sq. in. was used, this roughness could not be avoided. I have used our engine with a 5 to 1 ratio and found it snappy, and I have also used a 4 to 1 ratio with equal snap and obtained a decidedly smoother flow of power. With our cylinder, the amount of horsepower does not increase with the compression ratio above 4½ to 1. The volumetric efficiency decreases rapidly with an increase of the cylinder-head temperature resulting from a high compression-ratio. The lower compression-ratios can be made to give a higher volumetric efficiency, and therefore greater power from the same cylinder. The preignition point is the sum of the yoke temperature plus that of compression.

Our air-cooled cylinder will run satisfactorily up to a 630-deg. fahr. cylinder-head temperature, provided the sum of the yoke and the compression temperatures does not exceed that of the preignition point. The higher head-temperature will cool the engine at a greater horsepower on account of the increased temperature-difference between the cylinder and the cooling air.

Our present Series-10 powerplant has a compression ratio of 4.22 to 1, a compression pressure of 72 lb. per sq. in. at 400 r.p.m. and develops 24 hp. at 1800 r.p.m. with a cylinder-head temperature of 390 deg. fahr. when the cooling air enters at 85 deg. fahr. To obtain more horsepower, we would reduce the compression ratio and accomplish this result with the same bore and stroke.

The suction-yoke temperature on our Series-10 engine is now located at 160 deg. fahr. ± 5 deg. fahr. at which point we introduce a perfectly dry gas into the yoke. Should the yoke temperature rise to 180 deg. fahr. when the atmospheric temperature is 110 deg. fahr. in the shade, the engine would "ping"; should it drop to 110 deg. fahr., the smooth even flow of power would diminish in quantity and the economy would suffer.

The term, "compression ratio" is used frequently, and I often wonder if the public comprehends how misleading it is when discussed in connection with compressions or in connection with the number of miles per gallon. As an illustration, on wide-open throttle, I have measured an actual compression-pressure of only 60 lb. per sq. in. with a 5 to 1 ratio and have also measured a pressure

of 77 lb. per sq. in. with a 4 to 1 ratio. Both readings are true and consistent but, in themselves, they mean nothing unless something is known of the valve-timing, the yoke temperature and the manifold depression, as well as something of the port passages. To my mind, nothing could be more absurd than to use compression ratio as a basis for statements about miles per gallon on an automobile.

Our average touring car alone weighs 2450 lb. and an average five-passenger load weighs 750 lb.; therefore, the total running weight is 3200 lb. At a car speed of 20 m.p.h. at full throttle, the engine develops a brake mean-effective pressure of 75 lb. per sq. in. when using a mixture-ratio of 17 to 1, at a fuel-consumption rate of 0.607 lb. per hp-hr. At the same car speed, with just enough throttle to maintain this speed on a level road, the engine develops a brake mean-effective pressure of 16 lb. per sq. in. which, with a mixture-ratio of 13.65 to 1, is a fuel-consumption rate of 1.03 lb. per hp-hr. Under these conditions, the car traveled 34.2 miles per gal.

At full throttle, the pumping loss was based on a yoke suction of 1.05 in. of mercury. Under the condition of level-road operation, the pumping losses were increased in proportion to a yoke suction of 17.1 in. of mercury. The rolling friction of the engine does not change appreciably with load and, at 20 m.p.h., has a value of 8.5 lb. in terms of brake mean-effective pressure. The value of the increased pumping loss is $\frac{1}{2}$ (17.1 — 1.05) = 8 lb. per sq. in.

A brief analysis of the dissipated energy in the two cases stated at full load and at part load, is given in Table 3.

TABLE 3—ANALYSIS OF ENERGY DISSIPATED

	Full Load	Part Load
Brake mean effective pressure, lb. per sq. in.	75.0	16.0
Friction, lb. per sq. in.	8.5	8.5
Pumping loss, lb. per sq. in.	——	8.0
Total, lb. per sq. in.	83.5	32.5
Proportion of mean effective pressure generated that was utilized, per cent	90	49
Actual rate of power generation, lb. per hp. per hr.	0.546	0.505

Our company has endeavored to eliminate this waste by producing a car having a powerplant that will deliver 24 hp. at 40 m.p.h., when it knows that only 12.37 hp. is required to operate a two to five-passenger load on a level road. The car will travel 50 m.p.h. on the level. Were we to install a 45 to 80-hp. engine and build this car to endure, we would at once sacrifice its characteristics of light unsprung weight, remarkable fuel-economy, superior riding qualities and low maintenance cost. Our average stock car will accelerate with a full five-passenger load from 5 to 15 m.p.h. in less than 5 sec.; from 5 to 25 m.p.h. in less than 10 sec.; and from 5 to 35 m.p.h. in less than 15 sec. It can be throttled down to 3 m.p.h. in high gear, and it will run at a speed of 50 m.p.h. on a level road.

THE DISCUSSION

C. L. LAWRANCE:—As an aircraft-engine designer, the part of the paper that interests me most is the part dealing with engine performance. It is stated that the amount of horsepower does not increase with the compression ratio above a ratio of $4\frac{1}{2}$ to 1, and that an actual compression-ratio of 4.22 to 1 is used, which corresponds with good motor-car practice and, for many reasons, it probably is undesirable to use a higher compression than this. However, the fact that the amount of horsepower does not increase for compression ratios above $4\frac{1}{2}$ to 1 indicates that the cooling system is only just ample for the use for which the engine is intended; either the air is not sufficient in quantity or does not come into contact equally with all parts of the head, or perhaps the cooling area of the head is not sufficient. At any rate it is well known, especially in aircraft work, that it is possible to use successfully compression-ratios much higher than this.

The most interesting case of this is a recent experiment made by S. D. Heron at McCook Field while testing a cylinder of $4\frac{1}{2}$-in. bore and $5\frac{1}{2}$-in. stroke, having a cast-iron head and a combustion-chamber threaded on to a steel barrel. With a compression-ratio of 5.3 to 1, a brake mean-effective pressure of 130 lb. per sq. in. was consistently obtained, although the head temperature at certain points reached as high a value as 900 deg. fahr. It is fair to say that the head of this cylinder was considerably thicker than is the usual practice with cast iron, having been cast from a pattern that was intended for making aluminum cylinder-heads.

Our experience with various compressions leads us to believe that, all other things being equal, high compression with proper cooling will always give a better fuel-economy than a lower compression. Therefore, I believe that Mr. Grimes' illustration in which he compares two engines, one with a 5-to-1 compression ratio that shows only 60 lb. per sq. in. compression at open throttle, and another engine having a 4-to-1 ratio with 77-lb. per sq. in. compression, does not represent a fair comparison unless both engines are identical in every other respect.

I have just returned from Europe, and there is practically nothing to report with regard to air-cooled motor-car engines as no air-cooled European motor-car of the price and quality of the car Mr. Grimes represents is being built on the other side. However, I saw a large quantity of small cycle-cars being produced with the BSA engine that S. D. Heron described in his paper on Some Aspects of Air-Cooled Cylinder Design and Development.[2]

In England, a considerable amount of aircraft air-cooled engine development is in progress. The British Air Service is using a number of ABC Cosmos and Armstrong Siddeley air-cooled engines. The feeling there is that water-cooled aircraft-engines have seen their day, at least for most types of military aircraft. However, this was not true in France where, so far as I know, no air-cooled engines were in use. I saw an experimental one-cylinder air-cooled engine on test at the Royal Aircraft Establishment at Farnborough, that used the direct injection of the gasoline fuel. I was told that it showed a mean-effective pressure of 170 lb. per sq. in. This seems to indicate that the development of the four-cycle internal-combustion engine has by no means reached its limit.

C. P. GRIMES:—I have long known of your remarkable pioneer work in air-cooled engine construction. I know that you have been able to

(1) Build a lighter engine per horsepower

(2) Develop a greater horsepower per cubic inch of displacement

(3) Use less fuel per brake-horsepower hour

when using a compression ratio of 5 to 1 as against 4 to 1 with an aluminum air cooled cylinder.

We are not assured that our cars will be supplied with

[2] See THE JOURNAL, April, 1922, p. 231.

air plane high-test gasoline or benzol mixture. It actually requires 5.0 fan hp. to produce a cooling-air blast down past our fins of 90 m.p.h., which is the minimum velocity encountered by your engine mounted in the slip stream of your propeller.

Our engine must maintain its desirable characteristics of power, flexibility and economy over a period of from 10,000 to 15,000 miles of operation before overhauling any part of it. Ninety per cent of our entire mileage is covered at speeds of between 20 and 30 m.p.h. Our five-passenger car loaded requires 3.2 hp. at 20 and 7.2 hp. at 30 m.p.h. to roll it.

Suppose I did raise the compression to 5 to 1 and added a parasite resistance of from 3½ to 4 additional fan h.p. all for the sake of being able to operate this car for less than 10 per cent of its entire mileage with a compression ratio of 5 to 1 and a cooling air velocity of 90 m.p.h. not just when I pleased but only when I saw fit to spend 40 cents per gal. for airplane fuel and could afford to pay for a complete overhaul every few thousand miles.

If our Franklin car could fly or if it were to run on the top of Pikes Peak I would most assuredly make it with a 5 to 1-ratio with the full knowledge that the actual working compression pressures would then be practically the same as we now get with a 4.2 to 1 ratio at sea level.

Mr. Lawrance's statement "that high compression with proper cooling will always give better fuel economy than a low compression" does not agree with the facts in my automobile experience.

Ninety per cent of our car mileage is run on 20 per cent of the maximum engine power and requires an average of 5 b. hp. A cooling blast of 90 m.p.h. will require an average of 4 b. hp. additional just for the fan alone. Are we willing to increase the brake load 80 per cent for 90 per cent of the car life in the hope that a rise in the compression ratio from 4.22 to 1 to 5 to 1 will show sufficient economy in the use of 40-cent airplane gasoline to pay for this kind of an engine.

HERBERT CHASE:—What compression ratio is used and what actual compression pressure is realized at the speeds of maximum torque and of maximum power?

MR. GRIMES:—We use a 4.22 to 1 ratio. The maximum torque comes at 900 r.p.m. and a compression pressure of 70 lb. per sq. in. and the maximum power at 1800 r.p.m. and 65 lb. per sq. in.

H. SHEAFF:—Has Mr. Grimes any figure on the bearing pressure of duralumin, that he has obtained by tests?

MR. GRIMES:—No. It must have good lubrication and bear on a polished surface having a scleroscope hardness of 70.

J. G. PERRIN:—I have been deeply interested in the air-cooled-engine problem for some time. Two years ago I went to England to see what they had discovered during the war. Prof. A. H. Gibson read a paper on "Air-Cooling of Aircraft Engines," over there. He said they had discovered it wise to make the thickness of the cylinders greater than was ordinarily believed necessary. They found that the greater thickness conducted the heat better than the thinner section. It might be of help to know this in certain cases.

Has any work been done in adapting the Knight sleeve-type engines to air-cooled work? The Knight engine is used as a farm lighting engine now. Has there been any development or have any experiments been carried on regarding the possibilities of using a valve action such as sleeve-valves rather than poppet-valves in air-cooled engines? We all know that poppet-valves are very noisy in air-cooled engines due to the large clearances. With this forced air-cooling system as used by the Franklin Company can a Knight sleeve-type engine be cooled?

I believe that we are just on the verge of great developments in air-cooled automobile engines. So far, Mr. Grimes' company has tackled the problem in the most consistent way; it has built an air-cooled engine and then built an automobile around it. So many people have done the opposite. I believe that is a very fundamental point.

What has been the experience regarding the life of the cylinders in connection with the use of aluminum pistons? So many of us have had the experience that aluminum pistons lap the cylinders. With an air-cleaner, it seems to me that trouble will be solved. Has any difference been noticed in the life of the cylinders and the pistons, in connection with the use of an air-cleaning device?

MR. GRIMES:—We have observed that the thicker metal gave a better distribution of heat and in that way might aid its dissipation. However, in general we use the thinnest possible metal thickness because we find that the heat travels through faster.

Considerable work has been done by others with a Knight sleeve-type engine. Whereas, I do not believe that a double sleeve-valve engine can be developed with the same high efficiency as a poppet-valve engine, I do know that the Knight sleeve-valve engine can be adapted to air cooling if an aluminum cylinder-head is used. As a result of various tests with iron and aluminum cylinder-heads, I was given to understand that the engine would not be a success were it not for the use of the aluminum head that has capacity for taking away a large quantity of heat.

The Franklin cars have carried dust separators for the past 6 years. I feel that the separator on the Series 10 is far more efficient than any that we have used before. Piston life is very greatly affected by the care used in maintaining proper lubrication for the engine. We expect all of our pistons to last 15,000 miles and find a large percentage of them running 30,000 miles before it is advisable to remove them.

The difference in cleaner efficiency has not been sufficient to warrant my giving a definite statement of increased life.

H. M. CRANE:—It was brought out very clearly in this paper that an automobile today does not depend upon a particular method of cooling. Whether consciously or unconsciously, Mr. Grimes took fully half of his time in bringing out the excellent constructional features of the car, all of which, collectively and individually, have no bearing whatever on the method of cooling the cylinders. He is absolutely right. The public is interested in being moved from place to place, and it will not inquire too closely into how it is done, as long as it is done successfully, at a moderate cost and without too much trouble on its part.

I would like to repeat what I said at Dayton when S. D. Heron's excellent paper on some Aspects of Air-Cooled Cylinder Design and Development[3] was read. His paper, as well as this paper, are a challenge to the designer of water-cooled automobile-engines or water-cooled aircraft-engines. The water-cooled-engine designer has not been forced, until recently, to go into the facts surrounding his method of cooling. He has built a cylinder of a convenient mechanical shape, placed his valves where he wanted to place them and operated his piston in the way that he wished to operate it; then he has put a jacket on, poured some water in and said that

[3] See THE JOURNAL, April, 1922, p. 231.

it was properly cooled. The water might disagree with him. Frequently, the water does not go where the designer intends it to go. Few designers, until recently, have made any great effort to find out just what the water does in the jackets.

We had an analogous case on the Liberty engine, when we investigated it, with reference to the lubricating system. Some people concerned in the construction of the engine thought it was a good point that the engine did not require any oil radiator; in other words, that the lubricating oil that passed through the crankshaft did not get hot. The failures in those days were due to the fact that the crankshaft oil did not get hot; in other words it was not doing its part in cooling the crankshaft bearings. That was overcome by installing an oil pressure-lubricating system with a more active circulation. The same condition may be found in the cooling water of water-cooled cylinders. When corrected a larger radiator may be required, but the engine will run very much better.

Of the two types of engine, we may finally come to a division in point of size. The large air-cooled cylinders, while they have proved successful on test-blocks, have not given great success in practice so far. In aircraft engines, part of this is due to the head-resistance and to the great difficulty of controlling the valve-gear with reasonable weight and reasonable head-resistance. The expansion difficulty has been described and the ingenious method that Mr. Grimes' company uses for surmounting it.

Another feature of the air-cooled engine is that, in certain ways, it is far less accessible than the water-cooled engine. In its successful operation, it is practically limited to the valve-in-head type, with the valves integral and without detachable heads; in other words, any work to be done on a valve requires the lifting of a cylinder. The lifting of cylinders, in the hands of the casual owner, is a job not to be encouraged. The most difficult engine problem we have is to maintain the piston and ring seal in good condition; but the average owner, in lifting and replacing cylinders, is apt to make more trouble at this point than he will cure in the valves.

I have recently built an engine having an overhead camshaft, with an integral head carrying over the six cylinders but with the head detachable and the valves all in the head. We had it on the road only about 2 months before things occurred that made us want to look into the cylinders. We were not obliged to do this. The engine would operate and we could get over the ground with it, but there were certain things that we wanted to see. It took considerable determination to take the head off. I think that it ought not to be necessary to make access to the valves and the combustion chamber a difficult and tedious job or one requiring the disturbing of parts having no bearing on the work in question.

The tightness of an engine has much to do with its successful operation and even more with its economical operation. To encourage the operation of engines with leaky valves or leaky pipe connections, which I think might easily be the result of the slip-joints mentioned in the inlet-pipe design is not to the best interest of the automobile-using public. That is especially true in the larger engines. In the smaller ones, I think that we may find the field for air-cooling is widening all the time. It undoubtedly is widening and, if not, that will not be for the lack of the highest grade of research work, some results of which have been given in Mr. Grimes' paper.

MR. GRIMES:—I appreciate the remarks of Mr. Crane and will say that we feel our engine is far more accessible than the ordinary engine because we can remove any cylinder that we wish after the removal of nine nuts, each one of which is extremely accessible. I regret that Mr. Crane has not had an opportunity of becoming more familiar with our latest model engine, which is very accessible.

N. A. HOLLISTER:—The object of using the curved inlet to the fan wheel is not only to increase the capacity but it is also to increase the efficiency. Where only a small space is allowed the whole blade must be used effectively or the fan will not handle enough air. On the larger jobs such as those for big power companies where the power is a factor we increase the efficiency greatly by streamlining the inlet, such as has been done in connection with airplanes.

There was another way of increasing the efficiency of the Franklin car fan. We used to make a fan called a cone fan that was designed for discharging into space without a housing around it. We do not, however, get the efficiency with a cone fan or Sirocco fan discharging into space without a housing, that we do with a housed wheel such as used on the new Franklin model. The air will not be delivered as efficiently from an open wheel as it will if the wheel has a housing with the right scroll and the wheel is in the housing in the right position. These two points, the proper inlet and the proper housing design, are very important in fan design. I strongly suggest that the designer of an air cooled engine confer with the fan man, and as one speaker has mentioned design the engine with its cooling system and then build the car around it.

The importance of air cooling can be gaged by the many inquiries we have received, and we are working now on air-cooling problems for many companies. This air-cooling of engines, not only for automobiles but for other industrial usage, has aroused interest and I predict that it will not be many years before many trucks will be air cooled.

As to price, one company has a fan that is driven by a belt, which means another bearing and another piece of shafting. We can give them a wheel for less than the belts cost them, of course eliminating the extra bearing and, at the same time, handle more air with much less power than they use now. The elimination of freezing and other advantages appear on the face of it, and the fan men believe that the air cooling of engines is coming along rapidly.

MR. GRIMES:—The curved inlet on our fan enabled me to use a much shorter length of blade and thereby increased my overall efficiency.

R. B. WHITTINGHAM:—Was there any intention or expectation of deriving any flywheel effect in changing the position of the Sirocco fan from the rear to the front of the engine? Was that taken into consideration?

MR. GRIMES:—The flywheel effect is there, but the fan is most important. I am not willing to concede that the flywheel at the front end of the shaft is desirable in itself. We have it and our engine runs without producing a vibration period, but I credit that more to a 2-in. bearing on the crankshaft than I do to the flywheel on the shaft. I know of some shafts that had a number of vibration periods and they did place a flywheel at the front end of the shaft. It eliminated all the periods at lower speeds, but it had to be abandoned at the higher speeds. The fan on the front of our shaft is not there for the purpose of producing a flywheel effect, even though it is a flywheel.

C. T. MYERS:—What is the reason for using a 0.01-in.

tappet-clearance? Is it not possible to run with less tappen clearance, considering the compensating features that have now been adopted?

I note the hope of one speaker that truck engines will be air-cooled. I think there is a big field for the air-cooled truck-engine, but a campaign of education among the truck drivers is needed before they will be able to operate it successfully and satisfactorily.

I have found one trouble with the Model 9-B car; it will not pull very well at low speeds. This is undoubtedly due to the fact that the fan capacity is not sufficient at low speeds; there is not enough air passing over the cylinders. The car certainly drives very well on second speed on grades for a considerable length of time.

With reference to the dynamic balance of the crankshaft I believe Mr. Grimes stated that the flywheel is put into static balance; that the crankshaft is first put into static balance and then into dynamic balance with the flywheel attached; and that the fan was simply put into static balance. What about the clutch?

MR. GRIMES:—We use 0.010-in. tappet clearance for two reasons. First, our camshaft is designed to use this clearance quietly. We find that a little leeway is apt to allow the operator to run longer before adjusting it. Whereas, I am not personally familiar with the car Mr. Myers has been driving, we do know that in some cases the sheet-metal work under the hood has become bent and displaced for various reasons in such a way as to allow the circulating air to leak around the cylinder rather than to pass down through the fins and cool them. At the time of final dynamic balancing of the crankshaft, the flywheel and the clutch assembly are all checked when assembled into a single unit.

PROF. EDWARD D. THURSTON:—I understood Mr. Grimes to say that the quantity of air used is about 1 cu. ft. per sec. per hp. Is that independent of the engine speed? For example, the engine may be developing a maximum horsepower at rather a low speed, at which time the fan is delivering a rather small amount of air. Later, the engine may be delivering less horsepower, but at a very much higher speed; so, the amount of air per second per horsepower must be a function of the speed.

MR. GRIMES:—The volume of cooling air passed over our engine varies directly as the speed and is dependent upon the speed alone.

J. W. LORD:—Can you give us a comparison of the temperature of the newer engine with the older engine?

MR. GRIMES:—I have observed under similar dynamometer tests that our new engine reaches a temperature of about 380 deg. fahr. as against a possible temperature of 450 deg. fahr. for the Series-9 engine. These temperatures you must understand were taken at the hottest part of the combustion-chamber. Whereas, our former Series-9 engine would show heat on the road under trying conditions, you will find it almost impossible to observe any heat effect whatsoever on the Series-10.

SYDNEY G. TILDEN:—From an operating standpoint, most of us have experienced trouble with detonation. In a water-cooled car, it seems to me that the temperature-ranges of the cylinder and the cylinder-head between summer and winter can at least be controlled with radiator covers or some other such devices. With air-cooled cars the air that strikes the cylinder-head will be practically at the temperature of the outside air which will vary according to climatic conditions, say from 20 deg. below zero fahr. to perhaps 120 deg. fahr. in the shade. This would indicate to a casual observer that if the cylinder-head is properly cooled at ordinary temperatures it will be either over-cooled in extremely cold weather or under-cooled in extremely hot weather, which latter condition is likely to cause trouble from detonation or preignition.

It does not appear that the cylinder-head has much provision for cooling and that most of the cooling is carried along the side walls of the cylinder. How does Mr. Grimes control the temperature of the cylinder-head?

MR. GRIMES:—I have definitely determined by exhaustive tests that the temperature of the cylinder-head has nothing to do with the miles per gallon providing, of course, that the suction yoke-gases are delivered in a thoroughly heated and completely vaporized state as is now the case with our present construction.

I would call your attention to the fact that since the car takes but 3.2 hp. at 20 m.p.h. in the summer on a level road that it would not be difficult at all to consume far more horsepower than this in churning over the very heavy oils and grease that are sometimes left in the rear axle and transmission during very cold weather. Thick oil in the crankcase will not give proper lubrication to the cylinder walls. Our present cylinder design gives very satisfactory cooling indeed, even though the majority of the heat must be carried over to the fins and radiated from them.

The cylinder-head temperature is controlled by the gasoline mixture strength, by the compression pressure and by the volume of air passing to the cylinder-head. Our present design seems to have cared for detonation very nicely by eliminating it.

HAROLD L. POPE:—In aircraft work, it is necessary to get a very high mean effective pressure if it is expected to keep the weight per horsepower down to the minimum. I cannot see how the method used in the car under discussion would apply in aircraft work.

MR. GRIMES:—I will refer Mr. Pope to my reply to Mr. Lawrance and feel sure he will appreciate what I have said.

A number of questions have been asked regarding the cooling ability of our car, the amount of airflow and the like. I will simply say that our Series-10 car, with the five-passenger load and luggage, was driven from New York City to Oregon, then to Southern California, through Death Valley, through the Great American Desert and up Pikes Peak. It went far north into very cold country and it crossed the desert very successfully We have made tests repeatedly in which we have run the Series-10 car at 50 m.p.h., with wide-open throttle, up a long hill. Few hills are much over 1 to 2 miles long. We have a hill directly east of Syracuse that many cars cannot negotiate on high gear, yet we can take two people in our Series-10 phæton, that weighs 2450 lb. itself, and negotiate that hill on high gear at 50 m.p.h.

I am not thoroughly satisfied that cooling has nothing to do with preignition. With many water-cooled engines, preignition will occur if the compression-ratio is too high. The practical designer will design his water-cooled engine and will allow a little variation in the piston height, so that he can determine how the carbureter, the ports and the valve-timing work out; then he adjusts their relations so that the engine will work properly. That is all we do.

We have operated our engines from a 3.98-to-1 compression-ratio with our Series-9 car up to a 5 to 1 ratio with our experimental cars. They work well, but the 5-to-1 ratio is altogether too touchy and makes a snappy engine; as soon as a little is deposited, it produces detonation and, as soon as bad fuel is used, the engine is up against it; so, we use a compression of 4.2 to 1, which gives us accurate results with a low grade of fuel.

Our maximum torque does not occur at the maximum power. Our compression pressure for the maximum torque runs about 72 to 73 lb. per sq. in. We try to hold the compression uniform throughout the speed range. We find, by having a uniform compression-pressure throughout the speed range, that a given air-cooled cylinder will cool this uniformly and give us good results. Our maximum brake mean-effective pressure runs about 77 lb. per sq. in. Our phæton weighs 2450 lb. Its speed is 50 m.p.h. At 40 m.p.h. our car requires 12.47 hp. and it has 24 hp. It operates successfully with a wide-open throttle at 25 m.p.h. We believe we have covered the range that we have been asked to cover. Our plan is to try to have the power and the automobile correlated one to the other, so that we get riding power, fuel economy and riding quality all together, and a car that will run as fast as any ordinary person would expect to ride. An analysis of car operation is given in Table 3 of the paper.

Regarding the cooling effect from zero to 120 deg. fahr., in an effort to make a more economical automobile during the development stage, I tried the vacuum-controlled spark-advance, mentioned in the paper. It is well known that when running at part throttle, there is a high suction in the yoke and a low compression-pressure. People talk about high compression-ratio and economy. What good does it do, if you throttle the engine down and do not use it?

Concerning the volume, the cubic feet of cooling air per horsepower that flows over our engine and its relation to the speed, the volume varies almost directly as the engine speed. We cannot afford to change this one way or the other. It just happens to work out that our engine cools nicely at 450 r.p.m., at 50 m.p.h., and it takes about 550 cu. ft. per hr. of cooling air; whereas, for higher speeds, on account of the variation in the torque and the like, it works out to be 0.9 to 1.0 cu. ft. per sec. per hp. At the higher speeds, we could use still less cooling air; but, at the lower speeds, we need the cooling air.

In answer to the question on brake mean-effective pressure as compared to the compression pressure, a brake mean-effective pressure of about 77 lb. per sq. in. gives a compression pressure of about 70 to 71 lb. per sq. in.

I have been told that almost all the cooling in the Knight engine had to take place through the head; and that the oil-films and the sleeves were not good mediums to pass heat through.

We have used cylinder-heads 2 in. thick with perfect satisfaction. We went down to 3/8-in. thickness of cylinder-head but, when the head is too thin, the two valves in the cylinder-heads do not have a uniform distribution around the valve-seats and this is apt to give trouble. Therefore, we went back to a 5/8-in. thickness for cylinder-heads, and we have no trouble.

With an aluminum piston, we found that the cylinder life before regrinding ran between 25,000 and 30,000 miles before we applied the dust separator. Using the dust separator, that figure of 30,000 miles should be increased to 50,000 miles.

Arthur H. Holmes:—From the way Mr. Grimes' paper is written, it is reasonable to assume that it is his intention to encourage water-cooled engineers in the study and the investigation of air-cooling. Having this in mind, I suggest that Mr. Grimes differentiate clearly between those features of the car he describes that are essential to air-cooling and those that are used in its construction because the company thinks that they are proper and should be used whether the car had an air-cooled or a water-cooled engine.

As an illustration, the majority of the following items are not in ordinary use on water-cooled cars. A seven-bearing crankshaft; case-hardening of the crankshaft; a distributor oil-pump; the use of a special grade of oil; a double carbureter; a dust separator for air to the carbureter; an electric primer for starting; a temperature control on the intake-manifold; special pistons; and duralumin connecting-rods.

It seems to me that, if it is Mr. Grimes' desire to encourage interest in air-cooling, it would be wise to state definitely that the above special features of the car he represents are not used because they are necessary in connection with the air-cooling of an engine, but are simply features that his company believe are desirable with an air-cooled or water-cooled car. Certainly, in my experience, especially in some of the late developments that we have made we have proved it possible to build an air-cooled engine that is absolutely identical with a water-cooled engine in everything but the cooling system. For 2 years we have produced cars that will use satisfactorily any oil that a water-cooled engine will use, and use it with the same satisfaction that is secured with a water-cooled engine. In other words, the better the grade of oil is, the longer the life of the engine will be, but the question of a high flash-point or a low-carbon oil is no more important to an air-cooled than to a water-cooled engine. We have used a standard carbureter and a standard manifold with a hot-spot, and have obtained results comparable with those obtained in water-cooled cars.

We have been able to run an air-cooled engine at full load on the block without any external cooling system, ever since we used the aeroduct method of air-cooling, and we have used it for a little over 2 years, but I believe we are not able to do this at as high a speed as Mr. Grimes mentions. We have never actually tried it at more than 1500 r.p.m. of the engine but, up to that speed, it is perfectly practicable to operate at full power for any ordinary period. We have run up to 4 or 5-hr. tests and, at no time during these tests, did we even cool the oil by any external means. I judge that the amount of air passing by our cylinders is practically the same as the amount Mr. Grimes states as passing by the cylinders of the car he describes, but I am of the opinion that the power consumed is a little more than he mentions.

I think the big important point in connection with air-cooling that engineers must overcome before the use of air-cooled engines is extended to any great extent is the cost of construction. My judgment is that every air-cooled engine that has ever been produced so far has cost considerably more than the water-cooled engine. I am frank to say that there is no basic reason for this that I can see, aside from casting the cylinders individually, and I am not sure that this is absolutely necessary.

The impression could be created by Mr. Grimes' paper that the cost of the air-cooled engine in question is very high on account of the different special features that I have mentioned. Perhaps it would be well to state that, in a four-cylinder engine we have developed and got ready for production but have not yet started to build, the cost will be but little more than that of a water-cooled engine produced in the same volume. There undoubtedly will be a 10-per cent difference. There is absolutely nothing about this four-cylinder engine that is different from a water-cooled engine of the same size and construction, except the air-cooling and its effect on the cylinders. I think this is a very important point to have the water-cooled engineers understand. This four-cylinder engine is the best performing air-cooled engine I have ever used.

Its ability and its economy are especially good and, probably because of the simplicity of the structure, its reliability is equal to that of the best automobile practice of the day. Air-cooling should mean simplicity. I believe Mr. Grimes is just as anxious as I am that the water-cooled engineers should not get the impression that air-cooling demands a large number of special constructions that would interfere with simplicity.

Mr. Grimes:—I am pleased to agree with Mr. Holmes regarding the suggestion that the great number of almost essential refinements found in the Franklin products are not found in the water-cooled cars. At the same time I wish to call your attention to the fact that the exhibits at the Automobile Show this year would indicate the advisability of a seven-bearing crankshaft, the use of a dust separator and an electric primer for starting. I am inclined to predict that more of those features that are now found only in our product will be adopted by the other companies which in itself will be ample proof that these features are not essential to the success of air-cooling.

It has been the policy of the Franklin organization to cooperate with the owners especially in the question of oils to assure them the greatest satisfaction from the use of their car for every dollar expended for lubrication.

A number of my friends who are still using water-cooled cars have assured me that they have been well paid for their casual investigation of the Franklin products since it taught them the value of a good oil to their car. I wish to assure Mr. Holmes that the horsepower mentioned for moving our cooling air is correct to the best of our knowledge and belief. I might say to him that whereas he might not have felt justified in running his engine more than 1500 r.p.m. at open-throttle on the block, that I have just completed a very interesting little run on one of our engines that to the casual observer would be identical in every respect to our production and that I operated this engine with full throttle at 2500 r.p.m. for 1/2 hr. The room temperature was 90 deg. fahr., the cylinder-head temperature 254 deg. fahr. and the power developed at the end of the run was 33 hp. We maintained a stream of water on the under part of the oil-pan during this run which prevented the oil temperature from exceeding 150 deg. fahr. Otherwise the engine was cooled naturally by itself with the production fan.

COORDINATION OF MOTOR-TRUCK AND RAILROAD TRANSPORTATION

(*Concluded from p.* 124)

fact that this method seriously delays the traffic, and on the belief that the service is more expensive than dray or truck service. The remedy for this practice lies with the Interstate Commerce Commission, who should be petitioned to require the railroads either to discontinue performing a service of this character or to charge for it a rate commensurate with its cost.

A reason urged for retaining the trap-car service by those enjoying it is that it enables them to get better terms from draymen; that without it they would be at the mercy of the draymen. This implies that those who are without track location are mulcted by the draymen. This can hardly be the case.

Highway-Transportation Brains

I have tried to point out what seems to me to be the proper field for the motor truck and, in a general way, to indicate the necessity for further development along highway-transportation lines. It has been said that one of the reasons for the lack of development of highway transportation is the lack of properly organized highway-transportation companies, which should be in a position to operate over the highways along the same general lines as other forms of transportation. It has been said also that the lack of these organized companies is due in part to the lack of finances and to their not having the confidence of the public in the particular class of work they are trying to perform. This is true only in part. I believe that the lack of highway transportation is due principally to the lack of organized highway-transportation brains.

Motor-truck builders should undertake to cooperate closely with motor-truck operators. The motor-truck builders' responsibilities should not end when sales of their equipment are made. I believe that in the engineering forces now employed by motor-truck builders there are many men who should use their engineering experience along the lines of highway transportation.

I am of the opinion that it would be to the advantage of the railroads of the Country to include in their organizations men who have had experience in building and properly using motor trucks. The railroads, to their advantage, should arrange for a more liberal mixing of commercial and railroad brains. I believe that one of the causes that has had much to do with the present dissatisfaction on the part of the public is the fact that the carriers are trying to solve commercial problems with railroad brains. It is difficult for the average railroad man, who is trained along railroad lines, to view the problems of transportation along the same broad lines as does the man who has had a commercial training. A great many of our misunderstandings would disappear if the carriers would use more liberally the services of the commercially trained man. We shall not be able to secure the best results from our railroads until the commercial and the railroad brains of the Country get together.

Motor-truck builders might also to their advantage include in their organizations terminal engineers, who, with the experience gained while in the service of the rail carriers, should be able to teach the motor-truck builders some of the proper uses of the vehicle.

The public also will require further education along highway lines. Only when the railroad and the highway are properly coordinated will each find its proper place in the economic transportation problems of the Country. The proper understanding of the "field" of the various transportation agencies is vital, but in determining this field each form of transportation should bear its proper responsibilities and charges. No one class of common carrier should be subsidized by the Federal Government or the States to the disadvantage of the other competing agencies. Only by a mutual understanding of their proper relations can the good of the public as a whole be advanced. The encouraging of active and forced competition would in the end be destructive.

One Hundred Ton-Miles Per Gallon

By J. B. Fisher[1]

Mid-West Section Paper *Illustrated with* Charts and Diagrams

The two-fold purpose of the tests described was to acquire as many data as possible regarding the peculiar requirements of motorbuses, as viewed from the standpoint of power requirements and fuel economy, and to analyze the discrepancy found so often between the performance of an engine on the test block and the fuel economy obtained from the same engine under actual service conditions.

Following a general statement of conditions to be met, and an examination of the problems of the manufacturer as to why his choice of the various units and accessories is such a vital factor in fuel economy, the improvements accomplished are enumerated, together with the reasons and inclusive of the desirable and undesirable features of carbureter specification and miscellaneous factors.

The test equipment and methods are specified and discussed, the results obtained when using a steam cooling-system are presented and the general results are stated and commented upon.

Early in the spring of 1922, a series of tests was inaugurated at Waukesha, Wis., with a two-fold purpose in view. One was to acquire as many data as possible on the peculiar requirements of motorbuses, as viewed from the standpoint of power requirements and fuel economy; the other was to analyze the discrepancy that is found so often between the performance of an engine on the test block and the fuel economy obtained from the same engine under actual service conditions. Ideal conditions found in a laboratory do not prevail on the road but, with due allowance made for this, there is a wide gap between the ton-mileage of the conventional truck and what we felt could be obtained by a thorough analysis of the problem.

It has been stated that the potential energy in 1 gal. of gasoline is approximately 100,000,000 ft-lb. If we could use this energy with no thermal or mechanical losses, it would be sufficient to raise a 2000-lb. car vertically a distance of nearly 10 miles. Due to our crude methods of converting this energy into power, we are barely able to propel the same car twice this distance on the ground. If it were possible to eliminate the thermal losses from the engine, in spite of which we have reached our present figure of upward of 100 ton-miles, we would approach 400 ton-miles per gal. Obviously, it is not possible to eliminate such losses, but it is possible to locate the causes of many of them and, by reducing them to a minimum, to secure greater efficiency in the numerous units that exert such a pronounced influence upon fuel economy. As an example of this preventable waste, the fact was cited recently that if Ford cars gave as correspondingly good mileage as the Franklin, Dorris, Essex or several other well-known cars, the saving in fuel would be sufficient to purchase the Muscle Shoals plant every 18 days. The Ford case cited is perhaps extreme, but the factors that are responsible for this performance also affect the entire industry to a great degree.

To get a better understanding of this phase of the subject, let us examine the problems of the car or truck builder and why his choice of the various units and accessories is such a vital factor in fuel economy. After having determined the weight of his proposed job he decides upon an engine and, too often, he is influenced to choose too large an engine to meet the public demand for exceptional performance on direct gear; and this means relatively poor economy at the lower speeds at which the car will be operated 90 per cent of the time. As a rule, the foreign cars have much smaller engines and greater gear ratios than American cars, so that they can have the engine operating more often at a speed and a load that will give the best economy.

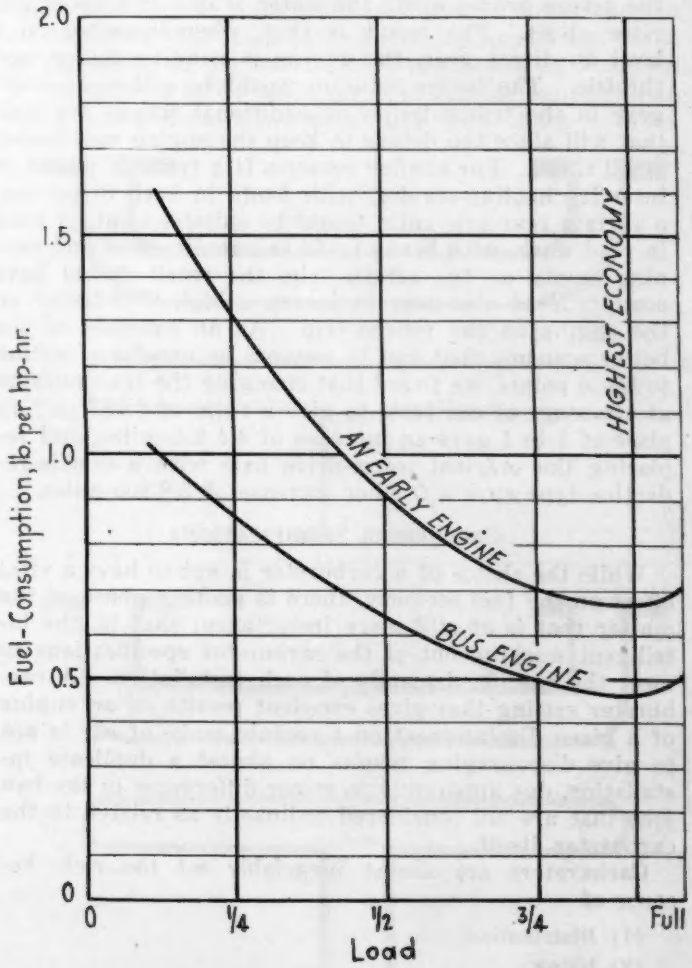

Fig. 1—Comparative Curves of the Fuel Consumption of an Engine Built a Few Years Ago and a Recent Motorbus Engine of Approximately the Same Displacement

[1] M.S.A.E.—Chief engineer, Waukesha Motor Co., Waukesha, Wis.

For several years, we have been studying carefully the factors that would let us get not only better efficiency at full load but, what is still more desirable, better economy at the light engine loads under which the engine often is operated most of the time. The results of this work are indicated in Fig. 1, which shows a fuel-economy curve from one of our earlier engines and one from one of our bus engines of about the same displacement. We have effected this improvement by

(1) Improved vaporization
(2) Use of higher water-jacket temperatures, without danger of hot-spots; cutting down losses to cylinder jackets
(3) Higher compressions, made possible by better distribution, improved cylinder-heads and the use of special aluminum pistons

The increase in thermal efficiency made possible by the use of the improved cylinder-heads and special pistons is dealt with in H. L. Horning's paper[2] on the Effect of Compression on Detonation and Detonation Control, printed elsewhere in this issue of THE JOURNAL.

After the decision in regard to the engine comes that of the transmission and the rear axle. They must be chosen to insure that the engine works as near full load and at as an efficient speed as possible at all times for, the nearer it works to full load, the greater the output of power per pound of fuel will be. Too often, a rear axle is chosen to permit the truck to meet some extreme conditions, as in pulling out of excavations or climbing the severe grades along the water fronts of seaport and river cities. The result is that, when operated on a level on direct gear, the engine is running under part throttle. The better solution would be either an overgear in the transmission or additional speeds available that will allow the driver to keep the engine well loaded at all times. For similar reasons, if a truck is placed in intercity hauling-service, with loads in both directions, a certain rear-axle ratio would be suitable; but, if used in road work, with heavy loads in one direction and running empty on the return trip, the truck should have some type of over-gear to insure a high load-factor on the engine on the return trip. As an example of the better economy that can be secured by careful attention to these points, we found that changing the transmission at one stage of our tests to give a ratio of 0.787 to 1 in place of 1 to 1 gave an increase of 4.4 ton-miles, and replacing the original worm-drive axle with a double-reduction type gave a further increase of 5.8 ton-miles.

CARBURETER SPECIFICATIONS

While the choice of a carbureter is apt to have a vital effect on the fuel economy, there is another phase of the matter that is of still more importance; that is, the intelligent working out of the carbureter specifications to meet the specific demands of each installation. A carbureter setting that gives excellent results on an engine of a given displacement on a certain make of car is apt to give discouraging results on almost a duplicate installation, due apparently to minor differences in the two jobs that are not considered ordinarily as related to the carbureter itself.

Carbureters are almost invariably set too rich, because of

(1) Distribution
(2) Idling
(3) Low water-jacket temperatures
(4) Starting and warming-up period in cold weather
(5) Low compressions in the cylinders due to part-load running

The mixture-ratios that the carbureter delivers at different throttle positions have a vital effect on economy. We have types of carbureter that

(1) Start with a rich mixture and, as speed increases, give a leaner mixture at all higher speeds
(2) Give a rich starting mixture, decrease the mixture-ratio and increase it again slightly at very high speeds
(3) Give a rich starting mixture and do not reduce the mixture-ratio sufficiently at any higher speeds, but continue to furnish an over-rich mixture throughout the range of the carbureter

MISCELLANEOUS FACTORS

The size of venturi tubes, fuel jets and bleeders must be determined under actual road conditions to give the best results; and the carbureter construction must be such that there will be a minimum variation from the original settings throughout the life of the car. That is, the working parts of the carbureter must be proportioned liberally to provide the proper assurance against wear, with the resultant fuel leakage and air leaks. Air leaks around the throttle shaft mean that the mixture must be enriched to compensate for the air that leaks in, especially under nearly closed throttle. The exceptionally long throttle-shaft bearings on Fifth Avenue Coach Co. buses are an example of the precautions needed to guard against fuel wastage at this point.

As supporting our promise that the identical carbureter settings that give good performance on the test block will, with the proper coordination of all other units, give equally good results on the road, we might mention that the carbureter used in these tests, after about 5000 miles of operation, was placed on a similar engine on the test block. With no change in the settings or adjustments, it gave a fuel economy of 0.51 lb. per hp-hr. with the engine operating a 20-in. fan as under service conditions.

A radiator and a fan that will cool an engine at full load in hot weather will just as certainly keep the water temperature too cool in cold weather, unless exceptional precautions are taken to correct this condition. With too cool water-jackets, we have high friction losses at the pistons and the rings, poor vaporization, dilution and low thermal efficiency. The relation of the fan to the radiator is important because, if it is not located properly, the corners of the radiator will be practically dead as the fan may consume considerable power in churning the air in the shroud.

Vacuum tanks sometimes draw the fuel over on the suction line, or deliver the fuel too rapidly to the carbureter.

Hot-air stoves, air-cleaners and the like are sometimes too restricted and do not allow an engine to attain its full volumetric efficiency.

Too often a cheap spark-plug is chosen, one that is damaged easily and gives a weak spark; or, one that, due to its construction, produces a spark knock and causes the operator to run with the spark too late for the best economy.

It does not seem at first thought that the way in which an engine is suspended in a chassis would affect the fuel economy; but, if it is not mounted properly, the weaving of the frame may distort a crankcase in this manner enough to cause the main bearings to be thrown out of line and actually stall the engine.

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[2] See page 144.

A magneto may contribute its share toward poor economy by furnishing a weak spark or by faulty action of the breaker mechanism at high speeds cause more or less missing.

Large bodies produce excessive wind resistance and, as the wind resistance is controlled mainly by the frontal area, it is important to be able to transport the maximum tonnage with a minimum frontal area.

This by no means limits the list of items that affect the fuel economy, although it comprises the major items that are passed upon, or often passed over, by the manufacturer. The above points are not cited in a spirit of criticism, but merely to illustrate the opportunities that are present for securing better performance and efficiency by a careful analysis of the relation of the various units that comprise the completed product. It is also well to mention that these same problems face the manufacturer who makes all of his major units in his own shop, as well as the smaller truck or car assembler; in fact, instances of neglect to recognize and correct the foregoing points are fully as common in the product of an organization of this type as in that of the assembler. When the truck passes into the hands of the owner, other factors appear such as the kind of fuel and oil used, attention paid to engine and carbureter adjustments, chassis lubrication and the like.

In addition to general conditions that confront the truck builder there are many other problems that, while related more or less to some of the above items, can be improved materially by recognizing these facts when designing the engine. To afford a better understanding of some of these problems, a brief description of the equipment used in this test is included.

Test Equipment and Methods

The truck used had a 2½-ton Sterling chassis, equipped with a large enclosed body having a frontal area of about 40 sq. ft. This type of body was used to let us approximate the bus type of body, as we were interested in getting as close to the conditions of bus service as possible. The tires were of solid rubber 36 x 4 in. front and 38 x 7 in. rear. The initial runs were made with a worm-drive axle that was used up to the last week or two of the tests. The radiator was of fin tubular type with seven rows of tubes; the area of the radiator was 560 sq. in. and the depth of core 3⅝ in. The fan was a Waukesha Motor Co. 20-in. four-blade, having a 2-in. projected width of blade, and running at about one and one-half times the engine speed.

Two routes were used for the test work. One of 24.4 miles was used for runs that would represent intercity traffic and another of 3.7 miles through the city of Waukesha to represent city running. The city run was made at different times with 12, 24 and 36 stops per trip. The last figure represents, approximately, 10 stops per mile. The load also was varied, from the 8300-lb. weight of the truck alone to a total weight of 16,000 lb. This was for the purpose of determining the effect of the various loads on the fuel economy. Most of the testing was done at a speed of 12 m.p.h. in the city and 15 m.p.h. in the country. This was varied at times but only to establish certain relations between speed and economy. A total of 296 observed runs was made, covering a period of a little more than 7 months.

At the same time these tests were being run we had another one of our engineers engaged in similar work in the congested districts of New York City, getting data for the Waukesha tests and, in turn, testing a number of the features brought out by our test runs. In the congested districts of New York City where this work was done, the average speed was sometimes as low as 5 m.p.h. with as high as 22 stops per mile. This would impose a very light load on the engine, which means that it would be operating down in the range of light loads and low speeds and, therefore, of poor economy.

The engine used at the start, at Waukesha, was our type CU 4⅜ x 5¾-in. unit, having a displacement of 346 cu. in. The improved thermal and mechanical efficiency of this engine, as a result of the better distribution and vaporization, together with the better performance obtained by proper coordination of the other units on the truck, made it possible to cut the size of the cylinder blocks from 4⅜ in. to 4 in.

One of the first problems that we undertook was to see what could be done to improve the carburetion. Several types of carbureter were tried. A carbureter that raised our initial mileage of 4.7 miles per gal. or 19.6 ton-miles to 5.5 miles per gal. or 22.8 ton-miles was finally selected for the test program. The representatives of the company making the original carbureter furnished with the truck were present at later stages of the test and, after going into the problem carefully, brought their performance in line with the mileage obtained on the carbureter that had been substituted for it to let us get somewhat more flexible carbureter settings.

We had observed, early in the tests, considerable condensation taking place in the manifold at the junction of the riser and the horizontal portion of the intake-manifold. O. H. Ensign, of the Ensign Carbureter Co., devoted considerable time in the past toward correcting the condensation that takes place at the tee and at the bends in manifolds of the conventional type, such as is shown at the top in Fig. 2. As a result of similar tests, we had for 3 years been using manifolds with sharp corners,

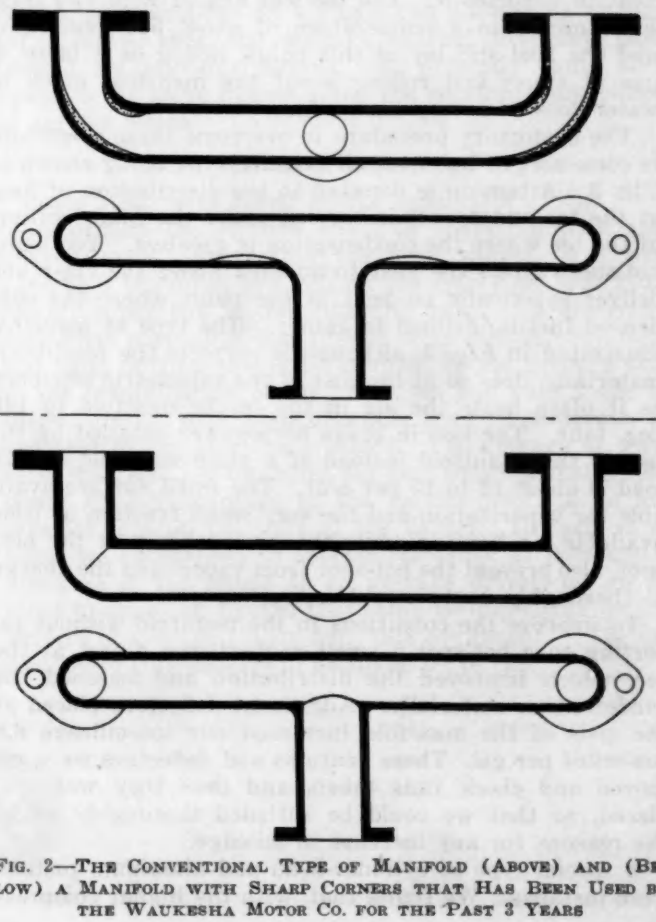

Fig. 2—The Conventional Type of Manifold (Above) and (Below) a Manifold with Sharp Corners that Has Been Used by the Waukesha Motor Co. for the Past 3 Years

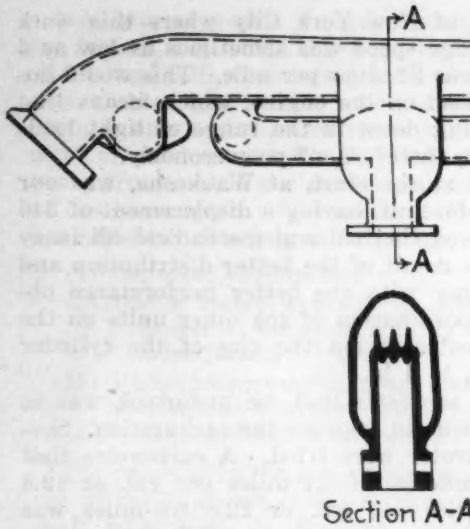

FIG. 3—A TYPE OF MANIFOLD THAT WHILE DISTRIBUTING THE HEAT SO AS TO ELIMINATE FUEL CONDENSATION OFTEN HEATS THE AIR UNDULY

such as shown at the bottom in Fig. 2. A small manifold with sharp bends may lessen the volumetric efficiency at high speeds, but the better carburetion and the improved performance at moderate speeds more than offset the slight loss in volumetric efficiency.

The tests described by W. S. James, of the Bureau of Standards, also showed the presence of large quantities of fuel at the tee in a Chalmers engine, when operated even under heavy loads, as indicative of similar conditions present in a great many other engines in popular use. He also showed in a glass manifold the tendency of the fuel to lie at this point, despite the application of heat to vaporize it. The tee was heated with two large blow-torches to a temperature of about 500 deg. fahr., and the fuel still lay at this point, riding on a layer of gas at times and rolling about the manifold much as water does on a very hot surface.

The customary procedure to overcome these conditions is some sort of hot-spot, an extreme type being shown in Fig. 3. Attention is directed to the distribution of heat at the tee and how it is carried under the inner corners of the tee where the condensation is greatest. Too many hot-spots direct the heat to an area above the riser and deliver practically no heat at the point where the condensed fuel is inclined to gather. The type of manifold illustrated in Fig. 3, although it corrects the conditions materially, does so at the cost of the volumetric efficiency as it often heats the air in the intake-manifold to 145 deg. fahr. The loss in brake horsepower entailed by the use of this manifold instead of a plain manifold at full load is about 12 to 15 per cent. The small surface available for vaporization and the very small fraction of time available for heating while the gas rushes past the hot-spot, also prevent the hot-spot from vaporizing the charge as thoroughly as we would desire.

To improve the conditions in the manifold without resorting to a hot-spot a small venturi was placed at the tee, which improved the distribution and lessened the condensation materially. Additional deflectors placed at the ends of the manifold increased our ton-mileage 8.8 ton-miles per gal. These venturis and deflectors were removed and check runs taken, and then they were replaced, so that we could be satisfied thoroughly as to the reasons for any increase in mileage.

A special type of cylinder-head and aluminum pistons were installed. We found that, with the higher compression that we were eventually able to carry with these heads, our mileage was increased 17.5 ton-miles per gal. With the higher compression we had a considerable surplus of power, being able to operate the truck at 45 m.p.h. and accelerate it from a standing start to 15 m.p.h. in 9 sec. This meant that, over a great portion of the course, the engine was under part throttle and operating at relatively poor thermal efficiency; so, to raise the load-factor on the engine, the bore was decreased from $4\frac{3}{8}$ to 4 in. We still had ample power and gained 3.6 ton-miles, due to operating the 4-in.-bore engine at a higher load-factor than the $4\frac{3}{8}$-in.-bore engine.

We had been running for several weeks without a fan, to hold our water temperature as high as possible. As a result of the better mechanical efficiency at the higher temperature and the slight saving of the horsepower consumed by the fan, we gained 4.5 ton-miles per gal. The temperature along the test route at this time ran from 80 to 90 deg. fahr. Although we could do this nicely in hot weather, we knew that it would be difficult to do so in cold weather; therefore, we began to search for a system of temperature regulation that would maintain the maximum permissible jacket temperatures under widely varying weather conditions and especially at idling speeds and light loads.

STEAM COOLING-SYSTEM

A steam cooling-system was installed to permit better control of the water-jacket temperature and to aid in giving still better vaporization within the cylinders in cold weather. As we had been able to hold the water temperatures at from 205 to 210 deg. fahr. we did not anticipate better economy, especially as we had been running for a month without a fan; and the use of steam cooling meant using a fan with a little additional

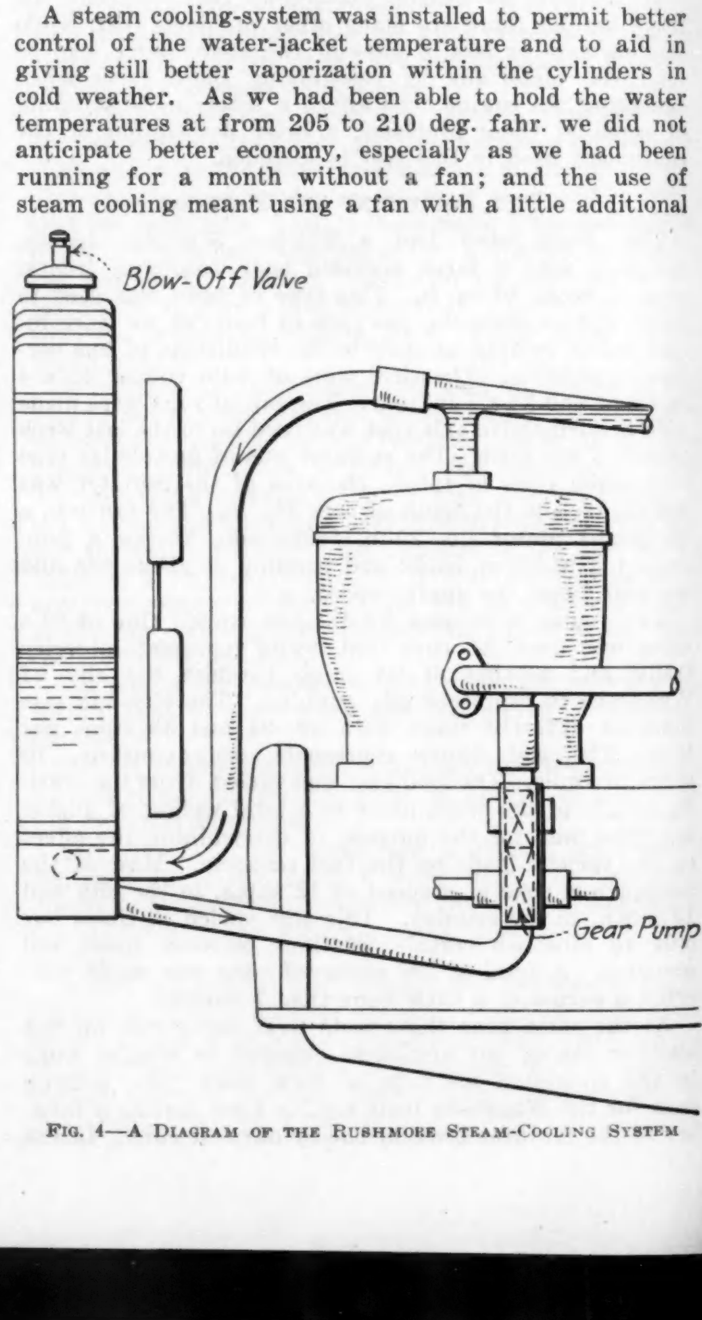

FIG. 4—A DIAGRAM OF THE RUSHMORE STEAM-COOLING SYSTEM

power to run it. There was no appreciable change in the fuel economy at that time with a 60 to 70-deg. fahr. air temperature but, with the advent of cold weather, we have been able to appreciate the advantage of this system. With a temperature of 2 deg. fahr. below zero, it is possible to hold a jacket temperature of 190 deg. fahr. with the same radiator as was used in summer, idling 4 hr. on 1 gal. of gasoline. The truck has been operated on the road at zero temperature with excellent results, with the identical carbureter settings used in July and August.

With a conventional water-cooling system, about 100 B.t.u. per hp. is delivered to the cooling water per minute on an L-head type of engine; or, in our case, about 3500 B.t.u. per min. With a 20-deg. fahr. temperature-drop, this would mean the pump would have to circulate about 24 gal. per min. Our pumps on this model are made to handle a little in excess of this to take care of constrictions in the radiator such as are found often after several months of service, and also meet the high temperatures encountered in tropical countries. In the early days, with tank cooling, this problem was badly misunderstood; some tractor builders seemed to think that if a large enough tank was used it would cool the engine, losing sight of the fact that, with no facilities for radiating the heat carried into the tank, the water in it would eventually boil if the engine were kept under load long enough. The early pioneers in the tractor industry were not the only ones to lose sight of the fact; in one of the largest engineering enterprises completed within the present generation, insufficient means were taken to radiate the immense amount of energy introduced hourly into the system, with the same results as were encountered early in the automotive industry.

A diagram of the Rushmore steam-cooling system is reproduced in Fig. 4. The top water-outlet on the engine, instead of being connected to the top of the radiator, is carried down and enters the lower tank a few inches from the point where water enters the line leading to the pump. A header pipe lies in the bottom of the lower tank and many small holes are drilled in this pipe to distribute the steam better throughout the width of the radiator. A gear pump is used in place of an ordinary centrifugal pump, as the water level in the engine when running must be maintained high enough to cause the water to flow over the top water-pipe. The gear pump is also more positive in action than a centrifugal pump and forces the steam upward and out of the cylinder blocks as fast as it forms. With a centrifugal pump, the formation of steam around the exhaust-valves or the spark-plugs, when once it is allowed to start, causes such a commotion that the normal flow of water is interfered with seriously, and it may even force the water downward and back through the pump. The gear pump serves as a traffic officer in this instance, with but one command to the stream of water and steam: "Keep moving and leave by the top exit." The actual amount of water moved by the gear pump on a steam-cooled job is far less than with water cooling, often but 20 per cent as much. The reason for this is that the temperature drop, even with the inlet water at 190 deg. fahr., is far greater than in a water-cooled engine, as the outlet temperature may reach 230 deg. fahr. There is also a greater temperature rise in the air passing through the radiator, due to the higher temperature of the contents of the radiator than in a water-cooled job.

The upper portion of the radiator is not filled with water, but serves as a condenser. If a shop steam-line is connected to the bottom of an ordinary radiator and a powerful electric fan is placed back of it, no steam will pass out of the top. If the fan is set oscillating, the steam will rush out in a 2-in. stream the instant the air blast leaves the radiator and stop the moment the fan starts the airstream again. With outlet-water temperatures of 225 deg. fahr., no knocking or evidence of hot-spots was found, but the engine ran decidedly smoother than with water-cooling and outlet temperatures of say 190 deg. fahr. The cylinder walls were hotter and of a more uniform temperature with the steam cooling-system, and the large area presented to the incoming charge as the piston was about to rise on the compression stroke was many times greater than the surface one could secure in a hot-spot on the intake manifold. The time element was also much longer when vaporization was secured in the cylinder itself.

A block test conducted several years ago with a type of rotating valve showed an appreciable increase of power and it was decided to install a set on the experimental truck and check the results of the block test. The rotating valves on the city running gave us an increase of about 2 ton-miles per gal., but on the country run the increase was 11 ton-miles per gal. The varying results that a given change would show on the city and the country runs was one of the interesting phases of the various changes made throughout the tests. The removal of the valve rotators on a check run showed no change for 3 days, then the mileage started dropping back toward its previous figures, but promptly came back again to the same figures when the rotators were placed in service again.

General Results

Some runs were taken at the conclusion of the tests to determine the relative effect of operating with still heavier loads. Increasing the total weight to 8 tons gave 89.6 ton-miles per gal. on the city runs and 115.3 ton-miles per gal. on the intercity runs. To determine just what portion of our results were due to equipment and what portion should be credited to the driver's skill in the manipulation of the truck, we had runs made by three different men, none of whom were truck drivers or had ever driven this truck before. The average of these three men was 110 ton-miles per gal., showing that a relatively small percentage of the results was due to any especial care on the part of the driver. In fact, at all times we endeavored to operate the truck as it would be operated in the hands of an average driver, as we desired to establish figures that could be duplicated by any drivers of average ability with this equipment. [The discussion of this paper will be found on p. 148.]

Effect of Compression on Detonation and Detonation Control

By H. L. Horning[1]

Mid-West Section Paper *Illustrated with* Charts and Drawings

SINCE the detonation tendency of the fuel is the limiting factor in the development of power and the efficiency with which the fuel can be burned, the author considers this phase of the subject with the idea of laying down the principles on which better economy can be attained through higher compression. The subject is discussed in regard to the causes of detonation and the methods of controlling them because detonation limits the compression at which an engine can run.

The phenomenon of detonation is analyzed, the author's opinion being that increasing the temperature causes an increasing frequency of radiant-energy impulses and that, finally, it reaches a point where the frequency corresponds with the critical rate of the electrons that bind the elements together; thus, it breaks them asunder and then the velocity attains the highest rate possible in a gas of that density and temperature.

The causes of detonation are enumerated and the methods of control are explained, consideration being given also to hot-spots, cooling difficulties and turbulence as controlling factors. A statement is made of the actual compression pressures attained without detonation in road tests, and charts showing the horsepower developed and the fuel consumption are presented.

AT present, gasoline is a drug on the market; however, the trend is toward scarcity. The huge stock dividends of the oil companies are an economic statement of the struggle to meet the demand on national resources; nothing, therefore, is so important as economy in consumption. A high duty is imposed on the automotive industry to set its house in order by producing the most effective apparatus for the conversion of the fuel energy into useful work, consistent with the psychology of the user and commercial considerations.

The general chassis conditions affecting economy, which are almost always neglected, have been stated; but I wish to emphasize them because we often find that the most economical engines, when applied to a vehicle, give a lower miles-per-gallon performance than a poor engine, because the units are not coordinated. Since the detonation tendency of the fuel is the limiting factor in the efficiency with which it can be burned and the power development, I have taken this phase of the subject with the idea of laying down the principles on which better economy can be attained through higher compression. Detonation limits the compression at which an engine can run and, therefore, we will discuss it in regard to its causes and the methods of controlling them.

For some 8 years American engineers have made an intense study of detonation, or what is commonly called "pink," "ping" and "knock." A dull thud, which is preignition, is due to a hot-spot in the combustion-chamber and is related only to the detonation because a detonation, if persistent, will end in preignition. But a detonation is a phenomenon related to that velocity of reaction which produces a sharp metallic sound from the cylinder-walls and, chemically speaking, is a reaction of such a high velocity that only part of the fuel enters into it; usually, the carbon fails to burn in the part of the mixture so affected. Physically speaking, when this knock occurs, the carbon particles glow and emit radiant energy. This is lost as useful energy and, further, it increases the temperature of the hottest spots in the cylinder, through the medium of carbon deposits. Thus, a vicious circle is established in which sections of the combustion-chamber become hotter under detonation. The cause and the effect unite to bring on the maximum effects, which are more violent detonation, preignition, further deposit of carbon and higher temperature, with the cycle repeating itself to the point where a heat balance is established temporarily, the gradual tendency being toward a hotter engine, until it is impossible to run the engine.

Many attempts to explain this phenomenon have been made considering the subject from the viewpoints of the chemist, the old school of physicists and the new school. My view is that increasing the temperature causes an increasing frequency of the radiant-energy impulses and that, finally, it reaches a point where the frequency corresponds with the critical rate of the electrons binding the elements together; thus, it breaks them asunder and then and there velocity attains the highest rate possible in a gas of that density and temperature.

Detonation Causes

We have held the opinion set forth by Thomas Midgley, Jr.[2] for some time; namely, that three factors control the weight of mixture consumed by the flame in a given time, in a uniform cross-section. These three factors are

(1) The reaction coefficient, which is characteristic of a fuel and is related entirely to the force diagram in which the electrons of the fuel are arranged
(2) Some power of the density of the fuel in the mixture
(3) Some power of the absolute temperature

Stated in every-day language, if a 53-deg. Baumé gasoline is tried, it probably will knock easily because it is composed of paraffines, which seem to be the least stable of all our common fuels; and, because it is so heavy, it will have large molecules that are also considered less stable than the lighter, simpler ones. The reaction coefficient, K, will then be very high. If a fuel of the Navy specification is used there will be less tendency to knock, but still it will be bad enough. With fuels of very high Baumé readings, the fuel will have less tendency to knock, due to its simpler structure. If olefines and naphthenes and aromatics are tried, the last being toluol,

[1] M.S.A.E.—General manager, Waukesha Motor Co., Waukesha, Wis.

[2] See The Journal, April, 1923, p. 367.

xylol, benzol and their derivatives, they will be found to be more and more stable and will not knock. The only difference in these fuels is the decreasing tendency to knock, so far as we are concerned with the running of the engine; all other conditions are practically the same, which is due to the reaction coefficient of the fuel as a chemical. Alcohol is among the most stable of all fuels and benzol has been run, by W. S. James and S. W. Sparrow, in a Liberty single-cylinder engine at a compression-ratio of 14 to 1 at the Bureau of Standards without detonation, as against our common ratio of 4 to 1.

Under the effects of density, all are familiar with the fact that when the mixture is lean, the compression low and the throttle closed, the engine will not knock. If we open the throttle wide, the engine may not knock until a certain speed, usually about 600 r.p.m., is attained. The engine will knock the richer the mixture becomes, until it becomes over-rich; then, the surplus fuel acts like any diluent or anti-knock compound and subdues the knock, very much at the cost of fuel economy. So far as the density is concerned, the more fuel there is in the air, at or below the perfect mixture-ratio, and the more weight of mixture there is in a cubic inch, the greater the tendency to knock will be. Usually, this is merely manifested to the user by knocking when the throttle is open, at the speeds of maximum volumetric efficiency and high load. At a higher speed, where the restriction through the valves hinders the fuel charge from entering, the weight per cubic inch of charge is less. So far as the engine design is concerned, the ratio of the volume when the piston is down to the volume when the charge is compressed is the fixed condition which primarily establishes the density under all conditions. Several factors determine the density of the charge, which, when it attains a certain critical value with a given reaction coefficient and temperature, will cause detonation.

Because there are a great many factors aside from heat that establish the density, and since the compression-ratio is the most important factor in economy, we must, therefore, suppress all other factors as far as possible so that the compression-ratio may be as large as possible. The following conditions affect the factor of density and must be held as low as possible:

(1) *Mixture-Ratio.*—The lowest ratio possible is controlled in a single-cylinder engine by the mixture-ratio at the spark-plug that will ignite promptly. In a multiple-cylinder engine, it is controlled by a ratio such that the mixture in the cylinder that gets the least mixture will ignite. This means, with bad distribution, that it is impossible to get low economies

(2) *Valve Opening.*—This must be as small as possible, so as to give a high velocity through the orifice to aid vaporization. This is true also of intake-manifold area; it is also true of the carbureter venturi-size which, together with the valve timing and the intake-gas temperature, determine the volumetric efficiency of the engine and hence its actual gage compression. With all these factors suppressed and the mixture-ratio as low as is practicable, the highest possible compression-ratio can be used

Speed of course affects the volumetric efficiency or the weight of the charge entering and, to have a high torque over a considerable range, it is necessary to have such a high charge-weight at full throttle and at slower speeds that the density is higher than the temperature conditions warrant to avoid detonation. Thus, to avoid detonation over a short range of slow speeds, the compression-ratio usually is reduced unduly. With the mixture-ratio at a minimum, with the temperature condition a minimum and with all other density factors held down, the compression-ratio can be maximum. This is the condition of the maximum indicated economy with a given fuel.

Friction horsepower and the pumping losses vary with the speed, and these are the final conditions that determine the economies at various loads and speeds. Aside from these conditions, a leaky valve and piston-ring combination shows a decided loss in the actual compression in the average engine. We have found it possible to increase the actual gage-compression 6 lb. by a better combination of piston and ring, and a further improvement of 3 to 4 lb. on the gage was obtained by securing a better seating of the valves. From these improvements alone, economy is certain.

It is commonly known that the engine will knock when the combustion-chamber is hot or when the air or the mixture is heated unduly before it enters the cylinder. We thus have considered all the factors that control detonation in view of our every-day experiences. Scientifically stated, they are: first, the reaction coefficient, which is a function of the structure of the fuel; second, the density, which is the weight per unit volume at the time the spark jumps; and third, the absolute temperature of the fuel at the moment of ignition, which determines its maximum temperature at all points in the cylinder and, for our purpose, the temperature of the

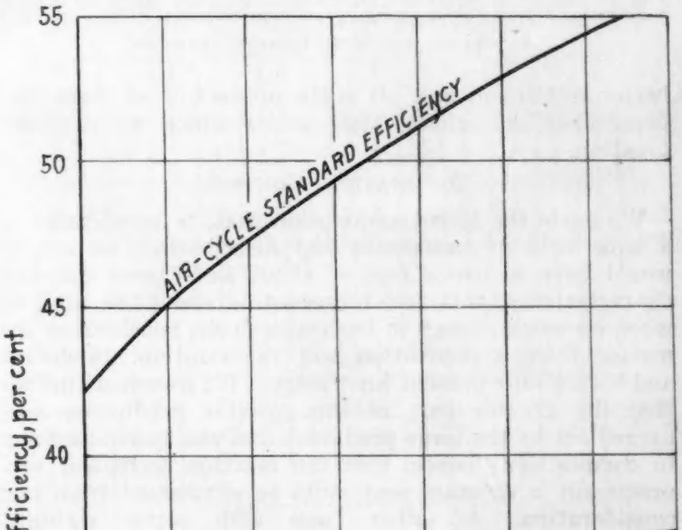

FIG. 1—CURVES SHOWING THE RELATION BETWEEN THE COMPRESSION-RATIO OF AN ENGINE AND ITS EFFICIENCY FOR THE AIR-CYCLE STANDARD EFFICIENCY AND THE OBSERVED THERMAL EFFICIENCY AT THE MOST ECONOMICAL MIXTURE STRENGTH

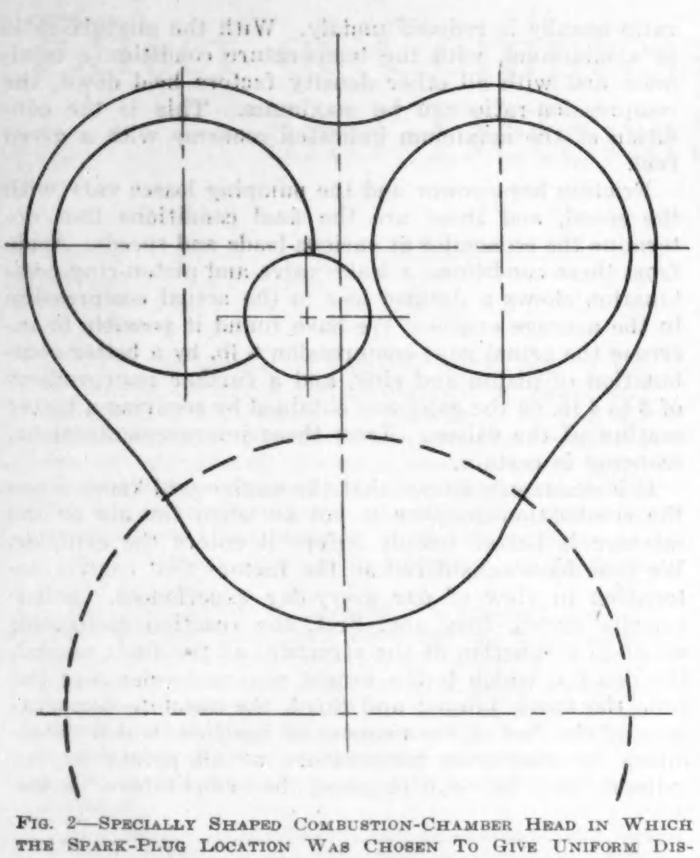

FIG. 2—SPECIALLY SHAPED COMBUSTION-CHAMBER HEAD IN WHICH THE SPARK-PLUG LOCATION WAS CHOSEN TO GIVE UNIFORM DISTANCES TO THE MOST REMOTE SECTIONS

hotter sections of gas. It is the product of all these that determines the critical state under which we have detonation.

DETONATION CONTROL

We made the broad assumption that, to be of value in a wide field of usefulness and distribution, an engine would have to use a fuel of about Red-Crown gasoline characteristics, this fuel representing about the limit of what we would expect to burn effectively, considering the matter from a detonation and vaporization standpoint and that of our present knowledge. We assumed further that the greater part of the gasoline production was turned out by the large producers and was fairly uniform in composition; hence, that the reaction coefficient was practically a constant and could be eliminated from our consideration. All other fuels with better stability would give better results.

Since the expansion-ratio determines one of the factors in the density, and therefore the efficiency of any engine, it is important that we examine very thoroughly the controllable causes of detonation that limit compression. Midgley's accomplishment is in controlling the reaction coefficient known as K in the equations used in physical chemistry; reducing it increases the compression, the temperature, or both. Our particular effort has been to discover what elements of design will subdue detonation effectively and thus allow an increase in compression, since this is the fundamental necessity for high economies. This brings us up to the relationship of those two factors.

As is well known, the efficiency of a heat engine increases with the expansion-ratio, sometimes called the compression-ratio or $R/1$, according to the upper curve in Fig. 1. This is known as the air-cycle efficiency. It is not possible to attain this efficiency in practice, because the actual working fluid varies from the ideal air assumed in the equation. Various deviations from the true cycle, losses due to changing specific heat of the working fluid, dissociation of the products of combustion and losses by radiated heat to the walls of the combustion-chamber, cause a further loss. Experience has taught us that these losses bring the actual efficiencies down to those of the lower curve in Fig. 1. Therefore, our problem becomes the simple one of designing a practical engine in which the cumulative effect of the heat of compression, the temperature of the incoming charge, the temperature of the products of combustion left in the cylinder and the temperature of the hottest areas in the cylinder allow a compression such that the engine operates just short of detonation under the most adverse practical conditions. Simply stated, if we can suppress the charge temperatures a little, we can increase the compression considerably.

Considering the above categories, we are entirely helpless to influence the heat of compression, as this is a physical characteristic inseparable from the condition. We are able to control the temperature of the incoming charge within certain limitations imposed by the design of the manifold, the volatility of the fuel and the temperature of the atmosphere. This is covered by J. B. Fisher, in his paper[3] One Hundred Ton-Miles per Gallon, and is a very important matter. Most attempts at improving the vaporization of our fuels that are low in volatility have been confined largely to heating the fuel either by first heating the air or by hot-spots located in the intake passages so as to be in the path of the liquid fuel gathered on the walls. Whatever has been the method, it has been difficult if not impossible to keep the sensible temperatures down and, invariably, whenever sufficient to affect the vaporization in the manifold appreciably, the incoming charge is so high in temperature as to force the use of unfavorably low compressions. The loss in thermal efficiency in these cases invariably is greater, when the compression is adapted to the high mixture-temperature, than the gain in the carburetion efficiency. At the high speeds and with the high throttle-openings, engines suffer a loss of power due to the low volumetric-efficiency imposed by the high intake-temperatures that are associated with hot-spot designs.

The intake gas-temperatures have a great influence on the maximum temperature attained in the explosion, and this is one of the determining factors in establishing the compression-ratio that is possible without detonation. The product of some power of the density of the charge and the same power of the absolute temperature must be within some maximum limiting amount.

Experiments, which are discussed in Mr. Fisher's paper, have proved that, as between a hot-spot system with a cool combustion-chamber and with a cold intake mixture and a uniformly hot combustion-chamber, the latter is preferable, from both the vaporization standpoint and from the standpoint of efficiency, as it allows a valuable increase in the compression.

This brings us to the temperature of the products of combustion left in the cylinder. This is affected largely by the initial temperatures, the completeness of combustion at the time of closing the exhaust and the temperatures of the hot-spots.

Henry M. Crane, E. A. Johnston and others frequently have stated their belief that the most effective place to vaporize is in the cylinder. This has been very apparent in all tests where a wide variation of intake temperatures did not improve the carburetion efficiency.

[3] See p. 139.

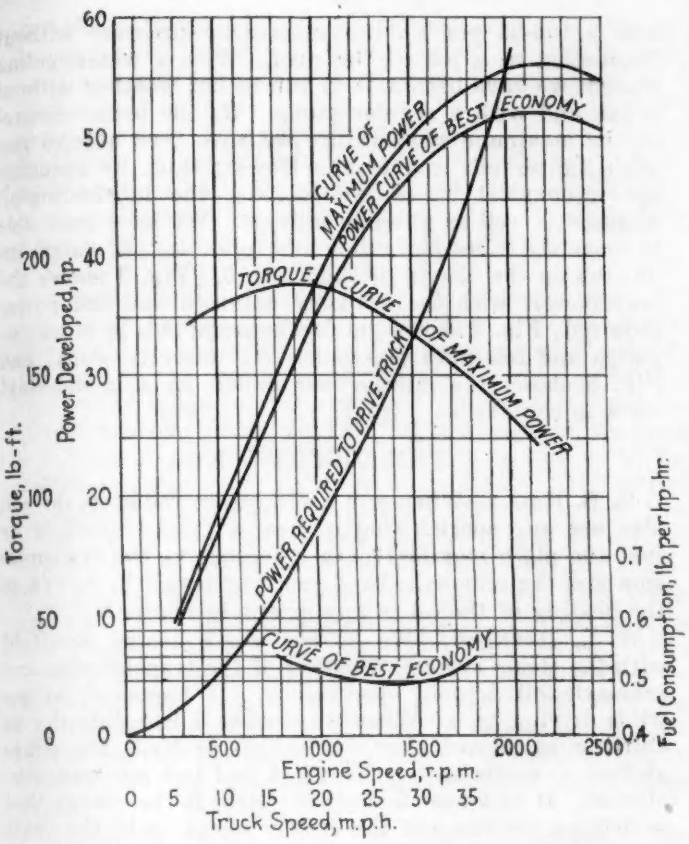

FIG. 3—CHART SHOWING THE POWER DEVELOPED, THE POWER REQUIRED TO DRIVE THE TRUCK AND THE FUEL ECONOMY OBTAINED

HOT-SPOTS AS CONTROL FACTORS

The hot-spots in the combustion-chamber are the most important conditions under the control of the engineer and offer an immense field for study and use. It has been pointed out many times before that the inherently high-temperature spots in a cylinder are as follows:

(1) First, the spark-plug electrodes, because of the adiabatic compression that surrounds the spark-plug during the entire duration of the explosion. Second, the unfavorable opportunity to cool its sections; that is, to cool the center electrode through the insulating material and the threaded sections of the spark-plug to the water-jacket, and due also to the distance of the central electrode from the cooling air. In connection with the spark-plug, we have found that there are a number of very good plugs that cool very well and give excellent account of themselves under practical conditions

(2) The exhaust-valve runs at red heat under high load, particularly if the seat is imperfect. It is through the seat that most of the heat is collected as well as dissipated. One of the most striking effects of turbulence is that, even though the exhaust-valve runs hot and has at times glowing particles on its surface, yet there may be and usually is no preignition because the gases are moving so rapidly over the surfaces of the valves. We have adopted a valve rotator so as to distribute the inequalities of the temperature and the surface and, by always seating in another place, reducing the permanent effects of local difficulties

(3) The temperature of the piston, particularly the piston center, the piston wall and the piston-ring temperatures, are very important elements in the operation of an engine. Mr. Fisher mentions these temperatures in his paper, in connection with their vaporizing effects and, as such, the piston presents the surface of the greatest value for that purpose, because of its area as well as its temperature. In connection with the piston, we have found it necessary to adopt aluminum pistons. We have followed the practice that was first adopted in aviation in this Country and in Europe: namely, a piston of the best alloy and ample head and ring sections to carry off the heat to and through the rings

As valuable as all these hot surfaces are as vaporizing elements, yet they often are too hot to effect vaporization because no liquid can spread on their surfaces at the temperature usually found. Any fuel that rests here assumes the spheroidal state and thus the vaporization is delayed greatly and is of questionable value. From a vaporization standpoint, therefore, it is of importance to hold these surfaces at a low relative temperature.

When it comes to the question of detonation, these surfaces are so disturbing as actually to control the question of the compression-ratio, other factors being the same. It is apparent that nothing short of the lowest possible temperatures of these parts, under the most severe conditions of running, is acceptable.

COOLING DIFFICULTIES

Aside from these well-known and well defined hot surfaces in the combustion-chamber, we must refer to some of the difficulties in cooling-water circulation that sometimes produce hot-spots. When there are pockets that give no easy escape of the rising water, or that cause

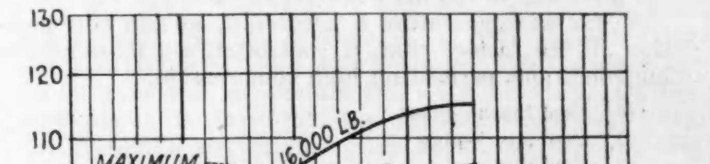

FIG. 4—FUEL-CONSUMPTION CURVES SHOWING THE MILEAGE AND THE TON-MILEAGE PER GALLON OF FUEL OBTAINED IN INTERCITY HAULING WITH THE TOTAL TRUCK WEIGHTS MARKED ON THE CURVES

steam to form from the lack of an unobstructed flow upward, then the water is forced from a hot-spot by belching steam and, when the water returns, the surface is so hot that it bounces off with little cooling effect. Thus a hot-spot develops and spreads and can be the cause of detonation; often, it is most difficult to find because there are no such distinctive marks on the cylinder-wall as appear when a piston or an exhaust-valve is too hot.

Where the steam system of cooling is in use, there appears to be a more uniform temperature of all parts; the maximum temperatures are lower and the average is higher and more uniform. This is exactly what is favorable for high compression. Hot intake-gases and hot-spots in the combustion-chamber are the curse of high compressions. The hot surfaces, presenting large areas to radiate heat, produce very sensitive volumes in the combustion-chamber which, when approached by the on-coming flame-crest, burn at the high velocities that characterize a detonation. Reducing the temperatures of these areas reduces the volume of these sections by drawing in the critical isothermal running over the hot surface, thus reducing the volume of mixture susceptible to detonation.

Turbulence

We have adopted a specially shaped modified Ricardo head of approximately the section shown in Fig. 2, which varies with the speed desired in a particular surface. In this, the spark-plug is located so that, as far as the large percentage of the mass of the mixture is concerned, it is in such a position as to be very close to all sections and the distances to the most remote sections are uniform.

So far as temperature is concerned, we can now mention all the factors that, if controlled, aid in establishing conditions permitting high compression.

(1) Cool intake gases
(2) Cool spark-plug
(3) Cool piston
(4) Cool exhaust-valves
(5) Relatively cool portions of combustion-chamber walls
(6) Turbulence

We have been able to reach a 5-to-1 compression-ratio

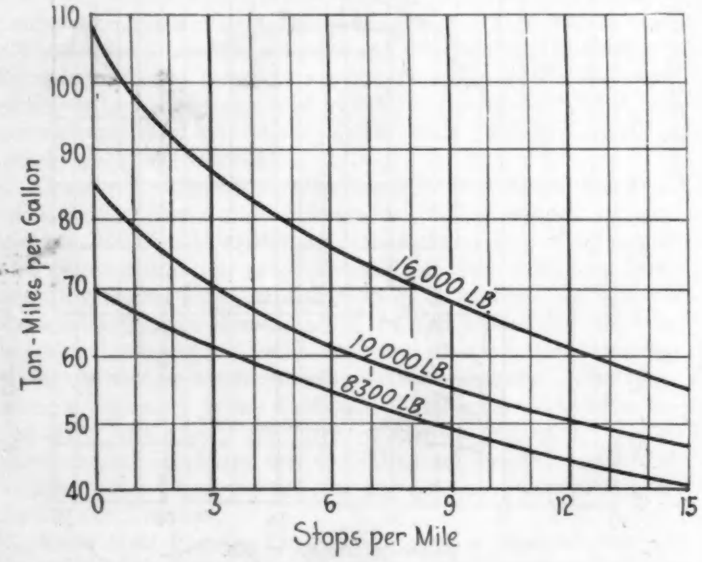

FIG. 5—FUEL-CONSUMPTION CURVES SHOWING THE TON-MILEAGE PER GALLON OBTAINED IN CITY TRAFFIC AT AN ACTUAL RUNNING SPEED OF 15 M.P.H. WITH THE TOTAL TRUCK WEIGHTS MARKED ON THE CURVES

and a 104-lb. per sq. in. compression-pressure without detonation in a job on the road. With a water-cooling system we have been able to run in hot weather without a fan and with a regular pump. By the better control of the maximum temperature we have been able to run with lighter oils and to run longer; thus, by reducing the mechanical losses, we increase the lubricating-oil mileage as well as gasoline mileage. We have been able to make the miles per gallon and ton-miles per gallon indicated on the charts in Figs 3 to 5. Fig. 3 shows the horsepower, with the economy obtained and the power required; Fig. 4 shows the fuel consumption in miles per gallon and ton-miles per gallon for intercity work; and Fig. 5 shows the mileage per gallon on stop-and-start work in city traffic.

THE DISCUSSION

R. B. HALL:—When you use a steam radiator, do you also use any special kind of heated intake-manifold or just the plain manifold? In reference to the transmission and the rear axle, have you ever tested in regard to the heating of the oil to any extent in winter?

H. L. HORNING:—We do not use a heated manifold with the steam system. Heating of the transmission and rear-axle oil actually does occur. It happened to me while driving here. When we started, I had difficulty in shifting to second gear; when we arrived, the gears shifted so easily that I thought I had lost my transmission oil. It took heat to do that. Heat is the energy that is driving the car and the energy which is in the shaft was converted into heat in the transmission and in the rear axle and then radiated; that heat goes out as absolutely lost energy. If you have these lubricants thin enough so that you do not need to heat them up to make them thin, mechanical losses are low. It should be remembered that viscosity is internal friction. Recently, in St. Louis, a test run between one-third and full speed showed a difference of from 3 to 8 hp. lost in the transmission itself. That was in warm weather.

J. G. ZIMMERMAN:—It is evident that the spark-plug threads do not carry the heat through very well. Has an experiment been made using plugs of larger diameter with a correspondingly larger area to carry the heat away? How about allowing the incoming air from the carbureter or the manifold to strike the spark-plug to cool it or as a means of helping to cool it?

MR. HORNING:—A body will radiate a certain amount of heat per square inch of surface at a given temperature. In a spark-plug, a piston or a combustion-chamber, the thing to strive for is a large volume with a relatively small surface. The section that we use over the piston is just as thin as we can make it because we find that, whatever we sacrifice in the volume of mixture we eliminate from the effective pressure because that volume burns too late and constitutes one of the sacrifices that we must make to use the combination. Using the incoming air to cool the spark-plug is an excellent suggestion; some combustion-chambers depend entirely on that. One in particular is called the pent-roof head. It has a head like the one shown in Fig. 6. When the intake opens the cool charge impinges on the spark-plug, and it does tend to keep it cool.

C. E. SARGENT:—These papers show the absolute necessity of looking after the small points. It seems that they have taken into consideration every little thing that helps in the efficiency and have taken care of it, which is a very good rule to follow when a high efficiency is desired. Mr. Horning stated that these little hot-spots

caused by the spark-plug and the valve and other things would prevent detonation if eliminated. How can they have any effect on detonation after inflammation has started? It seems to me that the elimination of the hot-spots would prevent premature ignition but that, after the inflammation has started, the hot-spots would have nothing to do with the detonation, the flame being very much hotter than the plugs themselves or any other hot-spot in the vicinity of the combustion-chamber.

MR. HORNING:—The particles of gas over a piston or any very hot sections are ready to detonate when the hot flames approach them. If you were to study piston temperatures and draw an isothermal line over it according to the temperatures of the piston, the isothermal line would bulge over the center. If we know the temperature above which we have detonation, then we have a volume of gas below this line which is sure to be involved in detonation under such conditions. The object in making a cool piston is to draw that isothermal line down so that we have a less volume of gas at critical temperatures. When the flame starting from the spark-plug approaches this volume, the ultra-violet rays reach out and set it off with a high velocity. The entire combustion-chamber is never involved in detonation. It is always some section like that, over the exhaust-valve or some uncooled spot in the water-jacket. In designing combustion-chambers, the spark-plug should be an equal distance from the hot-spots so that the flame will have traveled a distance that is a minimum to reach these points.

Surface-to-volume ratio is very important. Mr. Ricardo told me a year ago that he tested two engines of exactly the same size and exactly the same displacement. He found that there was absolutely no difference in horsepower and many other characteristics. Between a typical L-head engine and an overhead-valve engine at slower speeds, the heat loss was more at the moderate and at the higher speeds.

A MEMBER:—Mr. Horning made special mention of the importance of the engine accessories. Why do not the engine builders furnish the accessories that they deem advisable for a certain make of engine?

MR. HORNING:—The reason is that nobody will pay the engine builder a profit on them. Another reason is that the engine profits are so small that we cannot afford to send long distances to change defective accessories. There are so many kinds of accessories that the engine builder would be faced with an enormous inventory investment on which there was no profit.

A. Y. DODGE:—What is your analysis of the chief reason that you have been able to overcome carbonization? You laid stress on the importance of cooling the surface of the combustion-chamber; how do you keep this surface clean?

MR. HORNING:—The chief cause of carbonization is the carbon particles that glow during detonation but do not enter the combustion; when they get through glowing they drop dead, remain on the piston and all around it and cause trouble. If we prevent detonation, we can continue indefinitely without carbonization. If the piston is cool enough on the surface, the lubricating oil will not disintegrate. If we can keep the temperatures of the piston low, we can continue almost indefinitely without breaking down the oil or without this irregular combustion. Detonation is to a regular flame as a strike is to industry; a lot of energy is expended, but nothing but trouble is given back. Trouble is caused by the carbon deposited by detonation, and nothing is given in return for it.

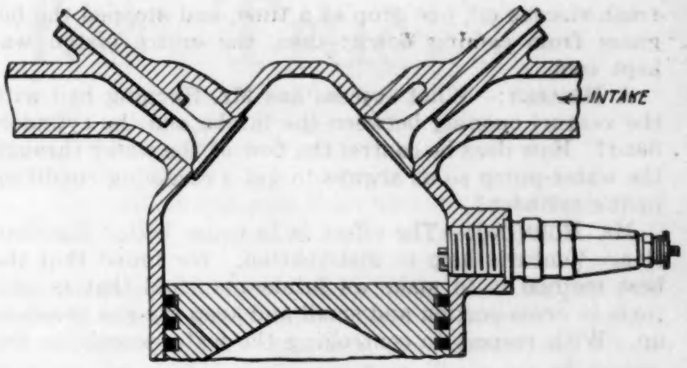

FIG. 6—A SO-CALLED PENT-ROOF TYPE OF COMBUSTION CHAMBER IN WHICH THE INCOMING AIR IS USED TO COOL THE SPARK-PLUG

W. S. HARLEY:—In speaking of piston-rings, Mr. Horning intimated that it is possible by design to get a better transmission of heat. To what sort of rings did he refer?

MR. HORNING:—It is not a matter relating to any particular ring but a matter of following the fundamentals. First, we must have enough metal in the head to conduct the heat entering the top of the piston to the surfaces of the ring on the groove side, which conducts the heat into the ring, and enough surface on the ring to conduct the heat over to the water-jacket through the oil film. We must have a balance between these. The important thing in piston and ring construction is the fit between them, because the heat path is through these surfaces.

A MEMBER:—Mr. Horning has given as the result of experiments that the heat would not pass through from one spark-plug screwed into another. The reason must be that there is an infinitesimal space between. On the other hand, he stated that the transmission of heat from the piston to the outside could come only through the piston-ring. There we have that space condition accentuated. Why should not heat pass through in one instance when it does in the other?

MR. HORNING:—You refer to the conduction of heat through a gas film between the exhaust-valve and the cylinder, where it was stated that a gas film is undesirable, as well as between the threads of a spark-plug and the wall into which it is screwed. In the spark-plug, unfortunately, we have a layer of gas between the threads, as is also the case between the exhaust-valve and the cylinder-wall. When we want a warm house we put an air space in the wall of the house. A thermos bottle is the best example of an air-gap. On the cylinder-wall we do not have a gaseous medium; we have a liquid medium which, we all know, is bad enough in regard to conducting heat. Lubricating oil is better than the gaseous medium, and that is the reason we have trouble in conducting heat away from the spark-plug and not from the piston to the cylinder-wall. Sometimes, when the piston is hot enough, the oil is so thin that the hot gases flow by and a vicious circle is established in which the piston gets hotter from the hot gases, the viscosity of the oil gets lower and more hot gases go by, and a gas gap and not a liquid gap prevents the escape of the piston heat; then the temperature rises until the engine stops from preignition.

Mr. Coleman, of the Madison Kipp Co., took one of our old engines that knocks badly, equipped it with his oiling system and stopped the knocking. The only reason the engine stopped knocking is that he succeeded in introducing an oil-film that had viscosity to it. He put in

fresh viscous oil, one drop at a time, and stopped the hot gases from coming down; thus, the entire piston was kept cool.

A MEMBER:—What success has Mr. Horning had with the venturi opening between the intake and the cylinder-head? How does he control the flow of the water through the water-pump so as always to get a steaming condition in the cylinder?

MR. HORNING:—The effect is to cause better distribution. Venturis help in distribution. We found that the best method is to make an intake-manifold that is uniform in cross-section and small and keep the gas speeded-up. With respect to controlling the water supply in the steam system, we must admit ignorance about that. We just put it on and it worked.

A MEMBER:—Is the combustion space immediately over the piston just a mechanical clearance? What has Mr. Horning found regarding the durability of aluminum pistons?

MR. HORNING:—I have answered the question about the clearance. If you get the section about ¼ in. or 5/16 in. thick, the detonation is very bad; if we make the section thin, the detonation will not be so bad. Aluminum pistons having a proper amount of metal and properly cooled appear to resist wear equally as well as iron pistons do when the vaporization is fairly good.

AMERICAN AERONAUTICAL SAFETY CODE

THE project for establishing a safety code for aeronautics has during the past year taken concrete form and its formulation is in active progress. The work is being pursued according to the scheme of procedure of the American Engineering Standards Committee which in 1920 recognized the Bureau of Standards and the Society of Automotive Engineers as joint sponsors for the project according to the rules of that committee governing work of this character.

A Sectional Committee to handle the technical work was formed during 1921 and at a meeting in New York City on Sept. 2, 1921, the permanent organization of this committee was effected, the officers being: Chairman, H. M. Crane, Society of Automotive Engineers; Vice-Chairman, Joseph S. Ames, National Advisory Committee for Aeronautics; Secretary, M. G. Lloyd, Bureau of Standards; Assistant Secretary, Arthur Halsted, Bureau of Standards. Five subcommittees were constituted, membership in which was subject to appointment by the Chairman and was not limited to members of the Sectional Committee. These subcommittees were appointed to deal respectively with the following:

(1) Airplane Structure, including design, construction and test
(2) Powerplants for Aircraft, including design, assembly and test
(3) Equipment and Maintenance of Airplanes in Service
(4) Lighter-than-Air Craft, including balloons, airships and parachutes.
(5) Airdromes and Traffic Rules, including signals and qualifications for pilots

The Sectional Committee is composed of representatives of the following organizations:

Aero Club of America
Aeronautical Chamber of Commerce
American Society for Testing Materials
American Institute of Electrical Engineers
American Society of Mechanical Engineers
American Society of Safety Engineers
Bureau of Standards
Forest Service
Manufacturers Aircraft Association
National Aeronautic Association
National Advisory Committee for Aeronautics
National Aircraft Underwriters Association
National Safety Council
Navy Department
Post Office Department
Rubber Association of America
Society of Automotive Engineers, Inc.
Underwriters' Laboratories, Inc.
United States Coast Guard
War Department
Weather Bureau

The work is intended as a cooperative research for the establishment of engineering safety standards in aeronautics, by which the relative merits or safety of aircraft, airdromes or their operation might be judged with uniformity by different individuals. The work was started with a synopsis of a safety code and preparatory draft made up from a study of rules and recommendations already in existence in aeronautical literature or compiled by various organizations for their own use. The subcommittees have studied and amplified this preparatory draft, modifying it to fit experience in practical aviation and supplementing it with material on subjects not already completely covered.

The American Aeronautical Safety Code will include various various parts as follows:

Introductory Part.—Scope and Nomenclature
Part 1.—Airplane Structure
Part 2.—Powerplants
Part 3.—Equipment and Maintenance of Airplanes
Part 4.—Signals
Part 5.—Airdromes and Airways
Part 6.—Traffic Rules
Part 7.—Qualifications for Pilots
Part 8.—Balloons
Part 9.—Airships
Part 10.—Parachutes

In the development of the code the broadest cooperation from the industry in general is desired and to secure this, parts of the safety code that are well advanced in the subcommittees are accepted as preliminary reports and copies circulated to persons who are sufficiently interested in the project to study and criticize them.

National uniformity in procedure and practice, where such is desirable, will undoubtedly be facilitated by this work and such uniformity can better be approached or established in the early stages of the art before diverse and conflicting local practices and laws become established.

The safety code can be used as a source of well considered information by agencies undertaking the establishment of airports and airways or commercial air services and will also be of assistance to the manufacturers of aircraft.

Three parts of the code, those relating to Airdromes and Airways, Balloons, and Airships, have been completed and submitted to the Sectional Committee, which has accepted them. These will be published in the near future and circulated among those interested in the development of aeronautics. They have not been adopted by the Sectional Committee and are subject to revision pending such adoption when all parts of the code are ready for consideration together. After adoption by the Sectional Committee, the code must be approved by the sponsor organizations before its promulgation.

In the Society the code will be submitted through the Aeronautic Division to the Standards Committee and the Society in accordance with regular Standards Committee procedure before its approvval by the American Engineering Standards Committee as a Tentative American Standard.

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Rear Axles for Trucks

By E. FAVARY[1]

METROPOLITAN SECTION PAPER

Illustrated with PHOTOGRAPHS AND DRAWINGS[2]

THE five types of final drive now in use on motor vehicles are stated by the author to be (*a*) the chain-and-sprocket, (*b*) the bevel-gear, (*c*) the worm-gear, (*d*) the double-reduction and (*e*) the internal-gear. The advantages of each type as emphasized by its maker are presented and commented upon, and the same procedure is followed with reference to their disadvantages.

Following these comparisons of the different drives, which cover about the first third of the paper, the bearing loads and shaft stresses of typical semi-floating and full-floating axles are calculated for the conditions (*a*) maximum torque plus the normal radial load on the wheel, (*b*) the wheel locked and skidding forward when the brakes are applied and (*c*) the wheel skidding sidewise while the truck is moving. A tabulation of the results obtained from the mathematical calculations is included. The author concludes from these results that while the maximum shaft-stresses are practically the same in both designs, the shaft in the full-floating axle can be made lighter and that a higher factor of safety should be employed in the semi-floating axle since the bending stresses are continually reversed. As the bearing loads in the full-floating axle are considerably higher, a greater bending moment is imposed on the axle housing, thus increasing the production cost of this axle because of the necessity for heavier bearings and axle housing.

The last third of the paper is profusely illustrated with photographs and drawings of various types of rear axle. These include the rear axle of the Class B truck, as well as commercial examples of the worm drive, internal-gear-driven axles with load-carrying members of different sections, double-reduction axles and an example of a chain final-drive.

IN all motor vehicles a certain gear-reduction, termed the final drive, takes place between the propeller shaft and the rear wheels. There are five general types of final drive in use: (*a*) the chain and sprocket; (*b*) the bevel-gear drive, with either straight or spiral gears, as mostly used on passenger cars; (*c*) the worm-gear drive, which is now the most common type used on trucks; (*d*) the double-reduction rear axle, in which two reductions are made between the propeller-shaft and the rear axle; and (*e*) the internal-gear drive, in which the first reduction is accomplished by bevel gears in the center of the axle, and the second by internal gears in drums or rings attached to the wheels. Each of the various final drives has its adherents, claiming advantages for certain types over those of others. I will endeavor to cite the advantages and disadvantages of each type as put forward in the literature of the various makers.

BEVEL-GEAR AND CHAIN DRIVES

In passenger cars the bevel-gear drive is employed almost universally, most companies using the spiral-bevel type on account of its greater silence in running and because it permits the employment of a smaller pinion. Since bevel gears are not, as a rule, practicable for reductions in excess of about 6 to 1, their use on trucks, except on the smaller models, is ordinarily out of the question, for here the final reductions vary from about 6½ to about 14, depending on the capacity of the motor truck and the truck speed.

The chain drive[3] on trucks is used on about 4 per cent[4] of the models manufactured. The chief objection to the use of chains is their exposure to dirt and grit, which in time causes excessive wear and noise. A number of attempts have been made to enclose the chains by covers of some kind, but they gave more or less trouble and were not entirely successful, for provision must be made in the cover for chain adjustment and for the relative lateral as well as the vertical movement of the sprocket centers on account of sidesway and spring flexure. Hence, today, most of the chain drives run with the chains exposed and by the employment of hardened chain-rollers, pins, bushings and links the wear and the noise are minimized.

Lack of lubrication is by far the most common reason for noisy chains. It is claimed that applying lubrication to the chains each day, which takes only a few minutes with a brush, results in greatly decreased noise and wear, and that with proper care and treatment chains will run from 40,000 to 50,000 miles. This figure is considered far too high by users of chain-driven trucks. The following are the principal advantages claimed for the chain drive over the other forms of final drive:

(1) Simplest in construction
(2) Will stand as much or more abuse than any other type
(3) Average total efficiency, including the losses in the jackshaft, is as high or higher than that of any other form of drive
(4) Increased losses due to worn chain and sprockets are very slight
(5) Easiest form of final drive to repair, as repairs can usually be made on the road with spare links
(6) Bevel and differential gears are carried on the sprung portion of the vehicle and are thus immune from road shocks
(7) A substantial load-carrying can be employed, which in the event of an accident will not damage the drive-shaft and the gears
(8) The differential gears and the drive-shaft can be made lighter than in most of the other types as the maximum torque is produced only at the rear wheels
(9) The universal-joints are arranged to run in line between the transmission and the engine, since both are attached rigidly to the chassis
(10) Offers the lowest unsprung weight, thus increasing the riding comfort

Some firms use the chain drive exclusively for their heavier models but give the purchaser the option of a chain drive or a double-reduction drive in the smaller

[1] M.S.A.E.—Consulting engineer, New York City.
[2] Copy for illustrations supplied by the McGraw-Hill Book Co., New York City.
[3] See TRANSACTIONS, vol. 9, part 1, p. 223 and vol. 11, part 2, p. 324.
[4] See *Automotive Industries*, Feb. 16, 1922, p. 362.

sizes. The former offers a greater road-clearance than the live-axle type of drive and a much greater range of gear-ratio changes than any other type. In the Mack truck, for instance, gear-ratios can be made from 6½ to 1 to 14 to 1, without altering anything but the sprockets and perhaps adding or taking out a link or two of the chain.

In addition to greater wear and noise, chains will require more frequent repairing and renewing than the gears of the worm or bevel drives. When sprockets are worn until they are hooked, or until the true form has been destroyed and the teeth are slightly hooked at the ends of the driving side, they will cling to the chain as it leaves the sprocket and are then pulled away violently. It is claimed that this can be remedied by some forms of sprocket design. The chain drive is superior to the worm drive under heavy load and torque and only slightly less efficient than the worm under high speeds and light loads.

The Internal-Gear Drive

In the internal-gear-driven axle,[5] first a bevel reduction is provided between the propeller-shaft and the rear-axle, as in most passenger cars, and to obtain a further reduction at the end of the live axle near the wheel or in the hub small spur gears mesh with internally cut gears attached to the wheel.

One of the advantages claimed for the internal-gear-driven axle is great strength, since it permits the use of a solid, one-piece load-carrying member, usually a drop forging of I-section or a round axle made from bar stock. Sometimes the load-carrying member is flattened and shaped in the center to encompass the differential housing, while in some designs the live axle is within the load-carrying member. Other advantages are (*a*) this axle can be made lighter than the worm-gear type, since the jackshaft and the differential run at a lower torque than in the worm-drive axle, the great speed-reduction being made at the wheel, while the reduction at the bevel gears in the center of the axle is usually less than 2 to 1; (*b*) the internal spur-gears and pinions are mounted on fixed centers and require no adjustment; and (*c*) the cost is low, while the efficiency is high and fairly constant at all loads.

One of the serious problems with most of the internal-gear-driven axles is the proper lubrication of all the parts of the drive, although it is claimed by its sponsors that this drive can run with very little lubrication. The large diameter at the ring gear gives plenty of opportunity for the grease to leak out around the edges of the gears and onto the brake. However, some types run in a constant bath of oil, and the joint is not close to the large diameter of the ring gear, but nearer the center. Another objection against the internal gear is that it is more difficult to provide the double internal-brakes on the wheel. Even where the internal brake is used, considerable difficulty is experienced with the rise in the temperature due to the application of the brakes, which may cause the lubricant to melt or burn. At present the tendency is to use internal rather than external brakes, as the latter form is exposed to dust and grit. The internal gear, as usually constructed, cannot be made in the semi-floating construction, thereby eliminating the use of this design where the bearing pressures can be considerably reduced over those in the full-floating type.

Opponents of the internal-gear axle further claim that it has several fundamental defects. In most of the internal-gear types the load-carrying member is the dead axle, but in some a central live member is inside the dead member, one or more extra idler gears being inserted between the jackshaft pinions and the internal-gear rings in the wheel-hub. In all internal-gear axles the internal gear is carried by the wheel, or else in the wheel-hub between bearings, and all wheels will develop a certain amount of play, since they have to take the road shocks. They will therefore not continue to run perfectly true, and the gears will not maintain their exact relation. If the gears run out of alignment with each other, instead of on a line contact, the contact will take place only on a point, under which condition gears cannot wear well or run silently nor transmit the power efficiently, for the slight play in the wheel will cause them to rock to and fro slightly.

In most internal-gear axles the wheels are mounted on some form of dead axle while the jackshaft, running parallel, is either in front of or behind it and encased in a separate housing, whose only function is to retain the live member. If they do not retain their parallelism, there will be an additional tendency to throw the gears out of alignment. The torque reaction tends to revolve the jackshaft about the dead axle and is restrained from doing so only by the security of the end fastenings and by the rigidity of the dead axle against twisting. If the torque were always equal at both ends of the jackshaft, the twisting moments would be balanced, but since the traction of the two wheels is not absolutely equal, varying greatly at times, the torque stress on the axle is not equal on both ends. It is therefore claimed by opponents that this inequality of torque upsets the absolute parallelism of the two members, which will have a tendency to throw the pinions out of alignment with the internal gears.

In some constructions the load is carried on the dead axle, the jackshaft being perfectly free, not centrally supported; hence it is not deflected when the axle is deflected, and this will have a tendency to pull the gears out of line and cramp the pinion bearings. Sometimes means are provided to take care of the deflection by providing a universal-joint in the jackshaft, as in the rear axle of the De Dion-Bouton Co., and the Rochet-Schneider Co. of France. This, on the other hand, introduces a more complicated mechanism and a larger number of parts.

The type that has a separate dead axle weighs more for a given strength, since the larger outside dimensions of a hollow tube offer more strength with less weight; hence for the same factor of safety a solid axle is heavier than one of tubular construction. However, against the disadvantages enumerated is the fact that about 17 per cent of the models of trucks built in this Country employ the internal-gear drive and among its adherents are some of the well known companies.

The Worm-Drive Axle

The worm and wormgear are usually assembled in a unit and then attached to the axle housing.[6] In this axle the road shocks or stresses, except those coming through the driving-axle, do not tend to disturb the alignment of the wormgear. Provisions should be made for preventing the oil in the center of the housing from running out through the hubs. Oil-retainers are sometimes used on both the axle-shafts and the wheel-hubs for this purpose. A vent pipe in the rear-axle housing will assist in overcoming oil leaks. Adherents of this axle claim that its average efficiency is very high and that a truck equipped with a high ratio will coast quite freely.

The great advantages claimed for the wormgear axle are the perfect mechanical enclosure of the final drive

[5] See Transactions, vol. 9, part 1, p. 207.
[6] See Transactions, vol. 9, part 1, p. 215.

which is running in a bath of oil, and its practically noiseless operation. Even the wheel bearings are lubricated from the center housing, and when proper provisions are made there is no danger of the lubricant saturating the brakes. When the housing is filled with a suitable oil, the latter will last up to 5000 miles. Due to the perfect enclosure, dirt and grit are positively prevented from reaching the mechanism. Worm-gear axles are frequently equipped with double internal-brakes, which affords excellent protection against dirt and grit. It is claimed that wheel brakes are more satisfactory than propeller-shaft brakes, for with the latter the stresses in the drive-shafts and the gears are considerably greater than when wheel brakes are employed, the torque arising from the locking of the wheels of loaded trucks being much higher than that of the engine.

Other advantages claimed for the wormgear are (*a*) its great simplicity, it having the fewest number of parts; (*b*) it is possible to obtain any desired reduction with two pieces, the worm and the wormwheel; (*c*) it is possible to obtain an almost straight drive from the engine to the rear axle; and (*d*) its durability is very great. It is better to have the weight in the center of the axle than at the axle ends, for when one wheel rises, or hits an obstruction and is thrust upward, the upward acceleration in the center of the axle will be only one-half of that at the wheel; hence the stresses induced will be smaller, and the action will be more like that of a smaller unsprung weight than if the weight were at the wheels.

Opponents of the wormgear claim that (*a*) it is a highly efficient power-transmission gear only under certain conditions, for under a heavy torque a wormgear has a very low efficiency, while at high speed under light load, its efficiency is high; and (*b*) in coasting there is a certain amount of friction in the worm and the gear. Tests made by the International Motor Co. showed that trucks equipped with a worm drive did not coast as well as other types; hence, it is claimed, that even though the worm may be more efficient than the chain under certain conditions, its average efficiency throughout the range of conditions encountered in service is lower.

The most fragile part of the worm drive is the worm thrust-bearing, which takes the enormous thrust exerted by the worm. In some axles ball bearings are provided at this point; others are equipped with taper roller bearings. Since the worm must be carried on the top to obtain sufficient road clearance, the worm bearings can be lubricated only by the oil carried up or thrown up as the wormwheel rotates. When this runs at a low speed, sufficient oil is not always carried to the top. To provide sufficient lubrication special means, like oil troughs, are sometimes provided inside the wormwheel housing to catch the oil and lead it to the bearings. The worm drive is more expensive than other forms, but it is claimed that the maintenance cost is considerably less. When the pressure on the worm is not too high, the gears properly cut and the lubrication adequate, the film of oil is not squeezed out from between the surfaces, and there is very little friction. However, when the pressure is excessive and the oil is squeezed out, considerable friction exists and damage may result in a short time.

Another reason for rear-bearing failures is that the necessary play in the worm takes the form of a blow on the thrust-bearing whenever the clutch is thrown in or speed-changes are made and, while taper roller-bearings or ball bearings can withstand high pressures, they cannot withstand impact very well. Trouble with thrust-bearings may also be caused by improper clearances, which, under excessive strain or deflection, cause an increase in the temperature with a consequent elongation of the worm. In the worm drive the torque induces pressure on comparatively small surfaces of the teeth, which rub one on the other, while with toothed gears or chains and sprockets there is more of a rolling contact than pure friction.

Some manufacturers claim that torque-arms and radius-rods impose additional stresses on the worm; hence they advocate the Hotchkiss drive. With this a cushioning effect is imparted to the drive as the springs permit the axle to rock back slightly in starting, thereby reducing the pressure on the worm teeth when the car is first started and the torque is highest. It is also claimed that it permits the worm and the wheel to oscillate, thereby working more oil between the surfaces than is possible with a rigid drive. On the other hand, opponents of the Hotchkiss drive in connection with wormgears claim that they must put extra weight into springs when the latter are used to perform the functions of the torque and radius-rods, and wherever satisfactory results are obtained with the Hotchkiss drive the springs must have a greater factor of safety. I believe that the Hotchkiss drive when properly designed gives satisfaction, even though the springs are relied upon to perform more severe functions. While many objections are brought forward against the worm drive, 72 per cent of all the truck models manufactured in the United States use this type of final drive.

The Double-Reduction Drive

In the double-reduction rear-axle two reductions take place between the propeller-shaft and the live axle. The first reduction is by a pair of bevel gears and the second by a pair of spur gears, or vice versa. Instead of straight-bevel or spur-gear teeth, helical teeth can be employed.

The advantages claimed for this type are (*a*) simplicity of construction, in that only comparatively small bevel and spur gears are employed, which lend themselves easily to quantity production; (*b*) all the gears are perfectly enclosed and protected from dust and grit, and run in a bath of oil; (*c*) its average efficiency is as high as that of any other drive and is substantially constant under all speeds and loads; and (*d*) the gears will always remain in alignment and are silent in running.

The disadvantages cited against this drive are (*a*) an increased number of parts over the wormgear drive; (*b*) it is heavier than the internal-gear and the chain drive, since the live axle has to carry the entire torque; and (*c*) it is more costly to manufacture.

Semi-Floating Axles

Having discussed the various types of final drive, before analyzing actual examples of the different designs we will calculate the bearing loads and the shaft stresses of semi-floating and full-floating axles, for with this information we will be in a position to determine their merits and demerits.

To find the bearing loads and the shaft stresses in semi-floating rear-axles three distinct conditions must be considered. These are

(1) The maximum torque plus the normal radial load on the wheel
(2) The wheel locked and skidding forward when the brakes are applied
(3) The wheel skidding sidewise while the truck is running

For example, a certain high-grade truck axle has a maximum total reduction of 49.61; the maximum horse-

power is 50 at 1000 r.p.m. and the load on each rear wheel is 9000 lb. The torque in the rear-axle shaft with a transmission efficiency of 85 per cent, and running in low gear is 132,850 lb-in. or one-half this amount, 66,425 lb-in., in each half of the rear-axle shaft. On the other hand, if a transmission brake is employed or if the engine is speeded up and the clutch thrown in suddenly, causing the rear wheels to slip on the ground, the maximum force at the periphery of the wheel is $0.6 \times 9000 = 5400$ lb., 0.6 being the coefficient of friction between the tire and the road surface, and if the wheel diameter is 36 in., the torque in each half of the shaft is $18 \times 5400 = 97,200$ lb-in. However, we will not consider these conditions, for in our example no propeller-shaft brake is employed, and steel can stand a large momentary occasional overload without serious damage. The dimensions of the axle in the before mentioned truck are 2 7/16 in. near the differential and 4 in. near the outer bearing.

The shearing or torsional stresses in the shaft can be found from the well-known formula

$$S_s = T/Z_p$$

where

S_s = the shearing stress in pounds per square inch
T = the torque
Z_p = the polar section modulus

The diameter at the weakest point of the axle, which is close to the inner bearing, is 2.4375 in. The polar section modulus at this point is 0.1963 times the cube of the diameter. Substituting these values in the formula for the shearing stress, we have

$$\begin{aligned} S_s &= 66{,}425 \div [0.1963 \times (2.4375)^3] \\ &= 66{,}425 \div 2.84 \\ &= 23{,}390 \text{ lb. per sq. in.} \end{aligned}$$

At the outer bearing the shaft is 4 in. in diameter and the stress due to torsion at this point calculated from the formula is approximately 5280 lb. per sq. in.

We will now consider the bearing loads and shaft stresses due to the bending moments. Under ordinary running conditions the radial load on each wheel is 9000 lb.; the distance between the bearing centers is 25 in., and between the outer bearing and the center of the wheel 6 in. approximately as shown in Fig. 1. If H_1 is the reaction or the radial load on the outer bearing, and H_2 that of the inner or the differential bearing, and if the radial load on the wheel is designated by P, then, taking moments about the outer bearing, we have $6P = 25H_2$, and $H_2 = 6P \div 25 = (6 \times 9000) \div 25 = 2160$ lb. Taking moments about the inner bearing, $31P = 25H_1$, and $H_1 = 31P \div 25 = (31 \times 9000) \div 25 = 11,160$ lb. The bending moment in the shaft near, or 1 in. from, the inner bearing, is 2160 lb-in., increasing toward the outer bearing, where it is $2160 \times 25 = 54,000$ lb-in.

The tensile and compressive stress S of a shaft is found from the formula

$$S = B/Z$$

where

B = the bending moment
S = the tensile and compressive stress of a shaft
Z = the section modulus

The section modulus, not the polar section modulus, of a round shaft is 0.098 times the cube of the diameter. Near the outer bearing of the shaft the diameter is 4 in. Substituting in the formula, we have

$$\begin{aligned} S &= 54{,}000 \div [0.098 \times (4)^3] \\ &= 54{,}000 \div 6.28 \\ &= 8600 \text{ lb. per sq. in.} \end{aligned}$$

Near the inner bearing the diameter is 2.4375 in. and by substitution in the formula we find the stress equals 1523 lb. per sq. in.

To find the stress in a shaft subjected to torsion and bending stresses, we may make use of an equivalent twisting moment T_e, which would create the same stress in the shaft as that due to the combined twisting moment T and the bending moment B and is equal to the square root of the sum of their squares. Near the inner bearing $T_e = \sqrt{[(2160)^2 + (66{,}425)^2]} = 66{,}500$ lb-in. approximately almost the same as that found for the twisting moment only. At the outer bearing
$T_e = \sqrt{[(54{,}000)^2 + (66{,}425)^2]} = 85{,}000$ lb-in. approximately
The stress in the shaft at the outer bearing is therefore

$$\begin{aligned} S_s &= T_e \div Z_p \\ &= 85{,}000 \div 12.56 \\ &= 6570 \text{ lb. per sq. in. approximately} \end{aligned}$$

While at the inner bearing $S_s = 66{,}500 \div 2.84 = 23{,}400$ lb. per sq. in. approximately, showing that the stress in the shaft is much greater near the inner bearing.

The material used for this shaft is S.A.E. Steel No. 2340, which, if heat-treated to a Brinell hardness of 335, or a scleroscope hardness of 51, has a tensile-strength of 175,000 lb. per sq. in. and an elastic-limit of 150,000 lb. per sq. in., according to the S.A.E. HANDBOOK. The shearing strength of steel is approximately 85 per cent of its tensile-strength or $175{,}000 \times 0.85 = 149{,}000$ lb. per sq. in., while the transverse elastic-limit or the elastic-limit in shear is approximately 35 per cent of its ultimate shearing-strength, and this should be considered when finding the factor of safety. Therefore, the elastic-limit in shear equals $149{,}000 \times 0.35 = 52{,}150$ lb. per sq. in. approximately. Very few data seem to be available as to the elastic-limit in shear of various steels. Hence, at the weakest portion of the shaft the factor of safety is $52{,}150 \div 23{,}400 = 2.23$ approximately under normal conditions, when running in low gear.

Next, we will investigate the bearing loads and shaft stresses when the brakes are applied and the wheel is locked and sliding forward. In this case there is a horizontal force at the periphery of the wheel equal to the vertical or normal load of 9000 lb. resting on it multiplied by the coefficient of friction 0.6, as was mentioned before, or 5400 lb. The two forces, the vertical and the horizontal, act at right angles to each other. The resultant radial load R on the wheel due to these two forces is equal to the square root of the sum of their squares or 10,500 lb. If we call the reactions or bearing pressures in the outer and inner bearings H_1 and H_2 respectively, by taking moments about the outer bearing we have $6R = 25H_2$, from which $H_2 = 6R \div 25 = (6 \times 10{,}500) \div 25 = 2520$ lb. Taking moments about the inner bearing, we have $31R = 25\ H_1$; hence, $H_1 = (31 \times 10{,}500) \div 25 = 13{,}000$ lb. The bending moment near the inner bearing is 2520 lb-in., and near the outer bearing it is 25×2520 or $6 \times 10{,}500 = 63{,}000$ lb-in. Then, since $S = B/Z$, at the inner bearing, $S = 2520 \div 1.418 = 1770$ lb. per sq. in., and at the outer bearing $S = 63{,}000 \div 6.28 = 10{,}000$ lb. per sq. in. approximately, the values for Z at the inner end of the shaft being 1.418 and at the outer end 6.28 as found before. It is seen, therefore, that under these conditions, with the wheels locked and sliding forward, the stresses in the shaft are very low, especially in view of the fact that the shaft is subjected to bending stresses only. The material is in tension and compression, and its elastic-limit is 150,000 lb. per sq. in.

We will now consider the bearing loads and shaft stresses resulting from skidding sidewise when turning a corner at a certain speed. Under these circumstances the centrifugal force F_c, which causes the rear of the

truck to skid, will equal the coefficient of friction, 0.6, multiplied by the total weight, $6W$, carried by the two rear wheels. In this case, therefore, $F_c = 0.6 \times 18{,}000 = 10{,}800$ lb.

Assuming that the center of gravity of the entire load on the rear wheels is located 45 in. from the ground, calling this distance h, that the total pressure between the two rear tires and the ground is designated by P, that P_o is the pressure on the outer wheel and P_i that on the inner wheel, when rounding a curve, that the tread, t, is taken as 56 in. and that the radius of the wheel, r, is 18 in., then, by taking moments about the point where the inner wheel touches the ground, we have

$$P_o \times t = (W \times 1/2t) + (F_c \times h)$$

or

$$P_o = 1/2W + [(F_c \times h) \div t]$$

Substituting the assumed numerical values, we obtain

$$\begin{aligned} P_o &= (18{,}000 \div 2) + [(10{,}800 \times 45) \div 56] \\ &= 9000 + 8700 \\ &= 17{,}700 \text{ lb.} \end{aligned}$$

The load on the inner wheel will be 9000 — 8700 or 300 lb.

In addition to the radial load, there is a thrust load L_t on the two wheels arising from the tendency to skid; this thrust load is equal to the centrifugal force F_c and is divided between the two rear wheels in proportion to their radial load. Hence, the thrust force on the outer rear-wheel between the tire and the ground $F_o = 0.6 \times P_o = 0.6 \times 17{,}700 = 10{,}600$ lb. Thus the total radial load on the outer wheel is 17,700 lb., and the side-thrust at the bottom of the wheel 10,600 lb. Taking moments about the inner bearing, we have

$$31P_o = 25H_1 + (r \times F_o)$$

or

$$H_1 = [31P_o - (r \times F_o)] \div 25$$

Substituting the corresponding numerical values for the various letters, we have

$$\begin{aligned} H_1 &= [(31 \times 17{,}700) - (18 \times 10{,}600)] \div 25 \\ &= 14{,}300 \text{ lb.} \end{aligned}$$

Taking moments about the outer bearing we have

$$\begin{aligned} 6\,P_o &= 25\,H_2 + (r \times F_o) \\ H_2 &= [6P_o - (r \times F_o)] \div 25 \\ &= [(6 \times 17{,}700) - (18 \times 10{,}600)] \div 25 \\ &= -3384 \text{ lb.} \end{aligned}$$

The result is a minus quantity as the bearing will carry the load on the top, while under normal running it will carry it at the bottom. It should be remembered that the outer bearing also has to carry a thrust load of 10,600 lb.

Knowing the bearing pressures or the reactions we can easily find the bending moments and the stresses in the shaft, as in the last example. The bending moment B near the inner bearing is 3384 lb-in. and near the outer bearing $25 \times 3384 = 84{,}600$ lb-in. approximately. In addition to the bending moment due to skidding, the twisting moment in the shaft due to the drive must be considered. Evidently the truck will have to be in the high gear to travel at the requisite speed to produce skidding; hence the torque will be much lower, but even if the truck were in the low gear, and the torque be 66,425 lb-in., the maximum stress in the shaft would be barely higher than under *straight forward* travel, when in the low gear. If the low-gear torque is added, the equivalent twisting moment near the inner bearing is practically the same as that found before, 66,500 lb-in., and the shearing stress equals 23,400 lb. per sq. in. Near the outer bearing the equivalent twisting moment has a value of approximately 107,000 lb-in., and the stress is 8550 lb. per sq. in. When skidding sidewise the torque is considerably lower in practice, for the truck would not skid unless it traveled in one of the higher-speed gears,

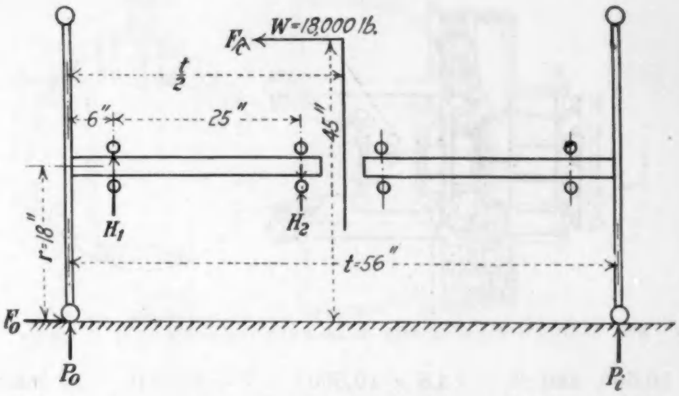


FIG. 1—DIAGRAM ILLUSTRATING THE BEARING LOADS AND SHAFT STRESSES IN A SEMI-FLOATING AXLE

unless the road surface were muddy or slippery, and in such event the coefficient of friction is considerably lower; hence the bending moment would be lower. The maximum stress in shear is therefore near the inner bearing on straight forward travel in the low gear and amounts to 23,400 lb. per sq. in.

FULL-FLOATING AXLES

The stresses in the shaft of a full-floating axle are purely torsional or shearing stresses, and if the engine power and gear reduction are as in the last example, the maximum torque will be as found before, 66,425 lb-in. in each axle-shaft. If the shaft dimensions are the same as those of the small end in the previous example, the maximum unit-stress for torsion in the shaft metal would be the same as that found before, 23,390 lb. per sq. in. In a full-floating axle the shaft dimensions are, as a rule, uniform, not tapered as in the semi-floating type, since the live axle is relieved of all bending moments by the wheel bearings.

The maximum radial load on the wheels, when the brake is applied and the wheels slide forward, is the resultant radial load due to the combined horizontal and vertical forces and is the same as found before, 10,500 lb. If the distance between the bearings is 7 in., as shown in Fig. 2, the inner bearing H_2 is 2½ in. from the wheel center, and the outer bearing H_1 4½ in., then, by taking moments about the outer bearing, we find $7H_2 = 4.5 \times$

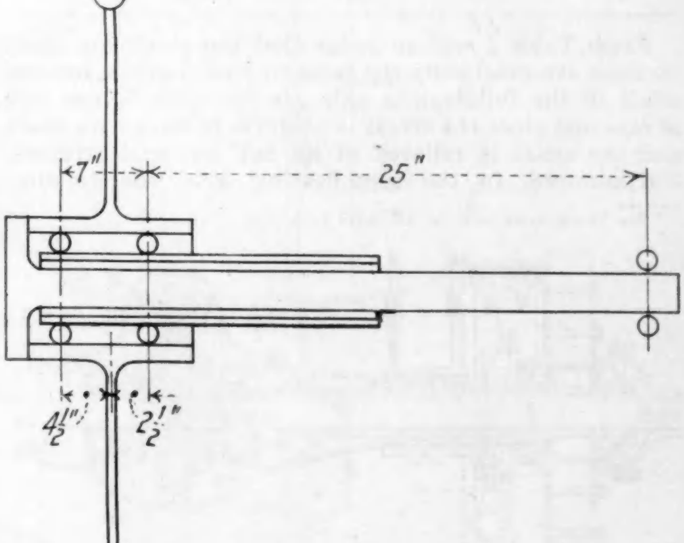


FIG. 2—DIAGRAM ILLUSTRATING THE BEARING LOADS AND SHAFT STRESSES IN A FULL-FLOATING AXLE

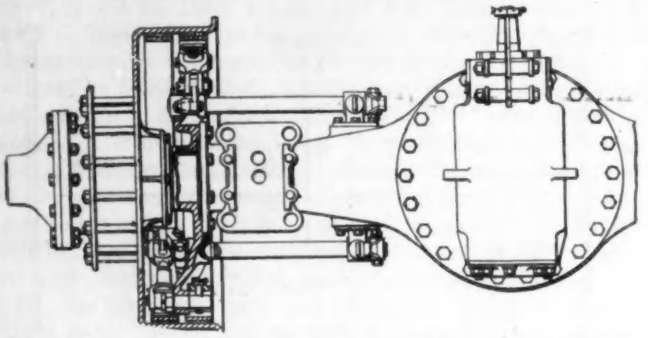


FIG. 3—PLAN VIEW OF THE REAR-AXLE FOR THE CLASS B TRUCK

10,500, and $H_2 = (4.5 \times 10{,}500) \div 7 = 6750$ lb.; the load on the outer bearing $H_1 = (2.5 \times 10{,}500) \div 7 = 3750$ lb.

When the wheels skid sidewise, the bearing loads on the outer wheel are considerably higher. If the portion of the centrifugal force F_c at the outer wheel that produces the skidding, and the radial load P_o are, as found before, 17,700 and 10,600 lb. respectively, then by the law of moments we obtain

$$7\,H_2 = 4.5\,P_o + 18\,F_c$$

hence

$$H_2 = [(4.5 \times 17{,}700) + (18 \times 10{,}600)] \div 7 = 38{,}700 \text{ lb.}$$

and

$$7\,H_1 + 18\,F_c = 2.5\,P_o$$

from which

$$H_1 = [(2.5 \times 17{,}700) - (18 \times 10{,}600)] \div 7 = -21{,}000 \text{ lb. approximately}$$

showing that on the outer bearing the load is reversed from its normal direction. In addition there is a thrust load of 10,600 lb. The results obtained are given in Table 1.

TABLE 1—COMPARISON OF STRESSES AND BEARING LOADS IN SEMI-FLOATING AND FULL-FLOATING AXLES

Type of Axle	Semi-Floating	Full-Floating
Maximum Stress, lb. per sq. in.		
Near wheel	8,550	23,390
Near differential	23,400	23,390
Maximum Bearing Load, lb.		
Outside of wheel	......	—21,000
Inside of wheel	14,300	38,800
At the differential	—3,384	......
Thrust load	10,600	10,600

From Table 1 we can judge that the maximum shaft stresses are practically the same in both designs, but the shaft in the full-floating axle can be made lighter and at less cost since the stress is uniform in the entire shaft and the shaft is relieved of all but torsional stresses. Furthermore, in the semi-floating axle, the bending stresses are continually reversed; therefore a higher factor of safety should be used. The bearing loads in the full-floating axle are considerably higher, which will impose a much greater bending moment on the axle housing. Hence the bearings and the axle housing must be made heavier and thus this axle will be more expensive to manufacture than the semi-floating axle.

[7] See TRANSACTIONS, vol. 13, part 1, p. 125.

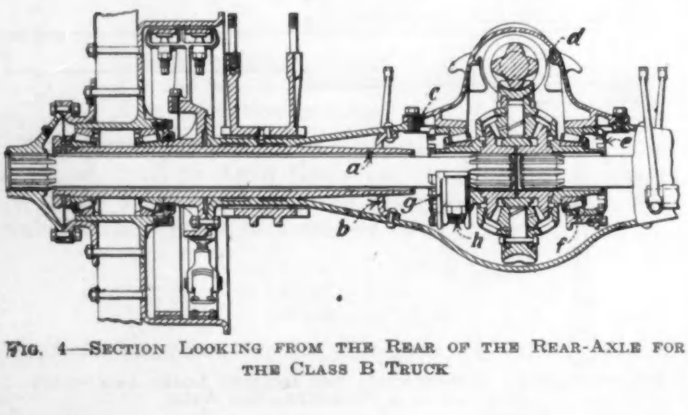


FIG. 4—SECTION LOOKING FROM THE REAR OF THE REAR-AXLE FOR THE CLASS B TRUCK

PRACTICAL EXAMPLES OF WORM-DRIVEN TRUCK REAR-AXLES

Figs. 3 and 4 give a plan view and a sectional view, looking from the rear, of the "Class B" rear axle. The housing is of the Timken type, with a square section from the center bowl outward. The tubes *a* are pressed into place, a retaining screw being provided in addition.[7] These tubes extend to the bowl of the housing. The reinforcing plate *b* is riveted to the housing and fits snugly over the end of the tube. The wormgear wheel is mounted on the differential by 97 splines, 1/16 x 3/16 in. section and 15/16 in. long. The differential, the worm and wormgear are mounted on the differential carrier, which is piloted at the top of the housing as shown at *c*, Fig. 4. At *d* are oil-grooves to lead the oil to the worm bearings. The means used for holding the worm bearings are shown in Fig. 5. The gear adjustment is made by slotted and threaded rings behind the bearing differential-cups, as shown at *e* in the assembly, Fig. 4. The roller-bearing cups *f* are clamped to the upper main differential-carrier by caps *g*, which are held by 1-in. nickel-steel studs *h*. The entire unit can be removed from the housing without disturbing any adjustments. The six roller bearings in the wheels and at each side of the differential are all identical in size and thus interchangeable.

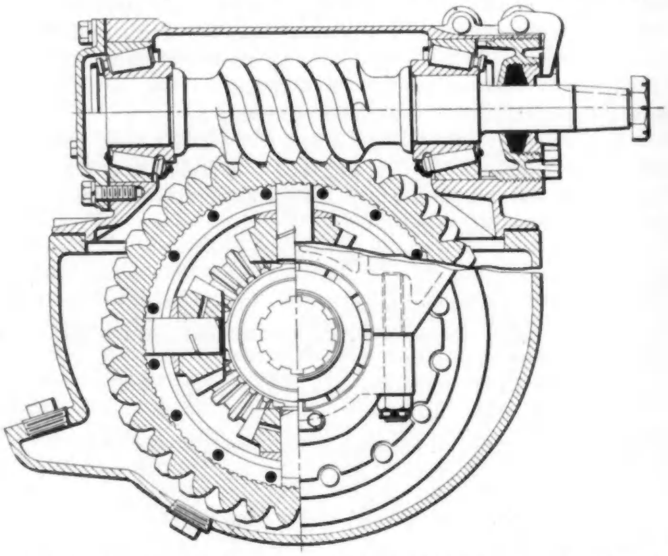


FIG. 5—DRAWING SHOWING THE ARRANGEMENT FOR HOLDING THE WORM BEARINGS

Fig. 6 shows a detail of the axle housing, which is a steel casting. The Government gave the manufacturers their choice of making this a steel casting as shown, or a steel stamping like that shown in Fig. 7.

Fig. 8 is the axle or drive-shaft made of chrome-nickel steel with a high carbon-content. At the weakest point the diameter is 2½ in. The axle is of the full-floating type; the radial load, as well as the lateral thrust, on the wheels is taken by taper roller-bearings, the shaft being only under torsional strain. The Class B truck is equipped with an engine having a torque of 2800 lb-in.; the ratio in the low gear of the transmission is 5.93 and in the rear axle 9.50; hence, the maximum ratio of re-

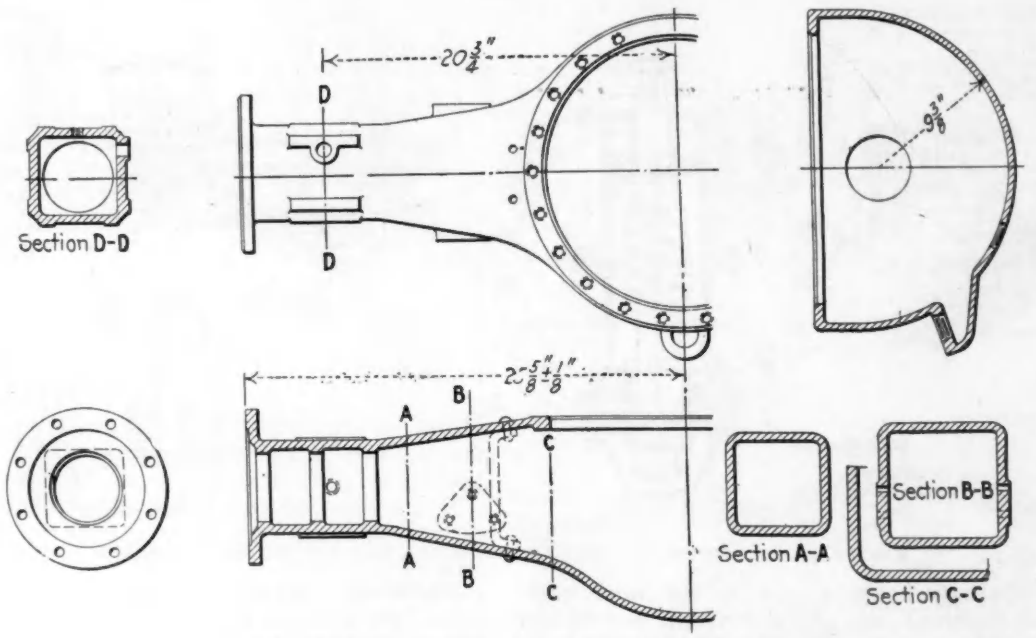


FIG. 6—DETAIL OF THE CAST STEEL AXLE HOUSING

duction is 56.3 to 1. The distance between the bearings in the rear wheel is approximately 5½ in.

Fig. 9 is the Sheldon rear-axle which is suitable for trucks of from 5 to 6-ton capacity. The maximum allowable load on the rear tires, including the weight of the truck and the pay-load, is 18,000 lb. The weight of this rear axle complete is approximately 2010 lb. In all the rear axles built by the Sheldon Axle & Spring Co. ball bearings are used for taking the worm thrust as well as the radial load of the worm, for which a number of advantages are claimed. The double-row bearing *i* at the rear carries the radial load as well as the thrust in both directions. It is claimed that with ball bearings no wedging is possible under thrust, whereas with taper roller-bearings such wedging exists. The front bearing *j* is free to move as the worm expands, due to a rise in temperature, which is occasioned by extra heavy work or lack of proper lubrication. No adjustments are necessary with ball bearings and, as the wormwheel and carrier are machined by very accurate jigs and fixtures, no adjustments are necessary in the first place. The semi-floating or fixed-hub type of axle is employed on all Sheldon axles; also the Hotchkiss drive.

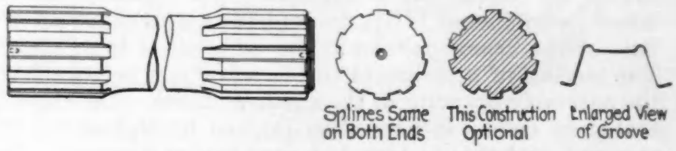


FIG. 8—AXLE OF THE CLASS B TRUCK

Fig. 10 shows the worm and wormwheel carrier, which can be removed from the axle housing as a unit complete with the worm, the wormgear and the differential. The lower section of the wormwheel runs in a bath of oil in

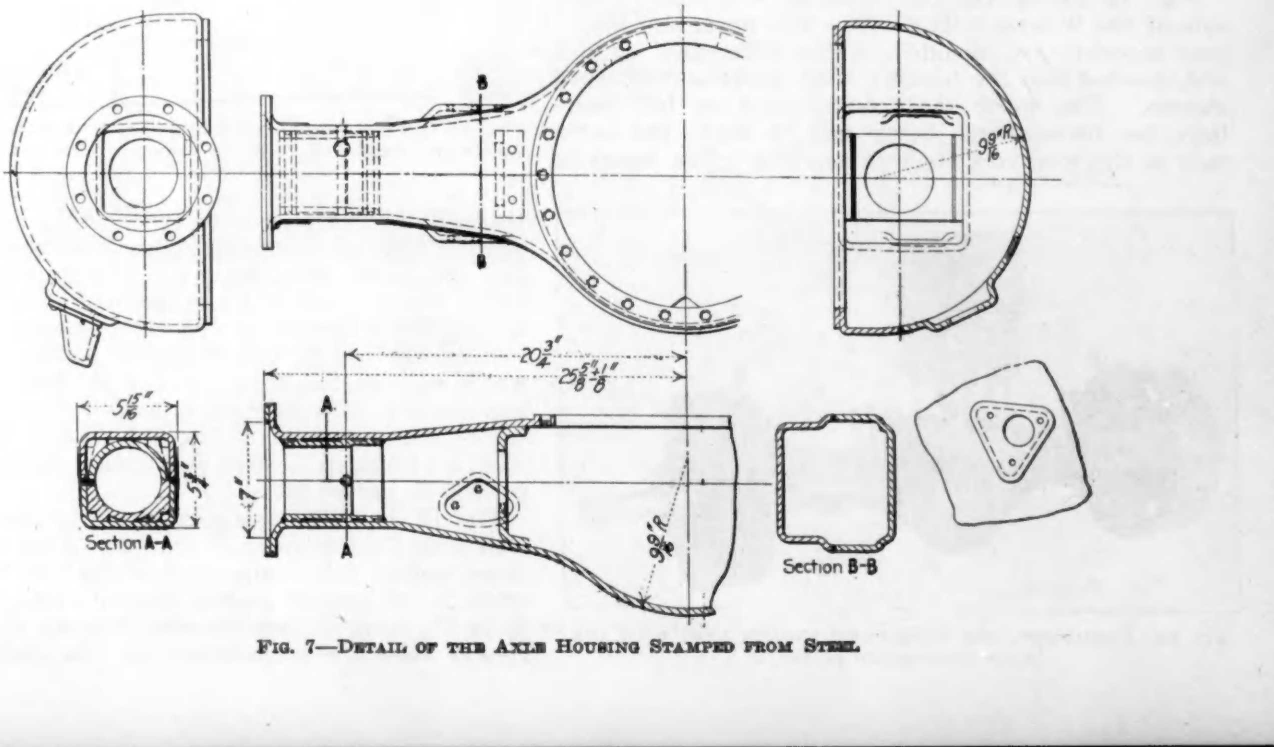


FIG. 7—DETAIL OF THE AXLE HOUSING STAMPED FROM STEEL

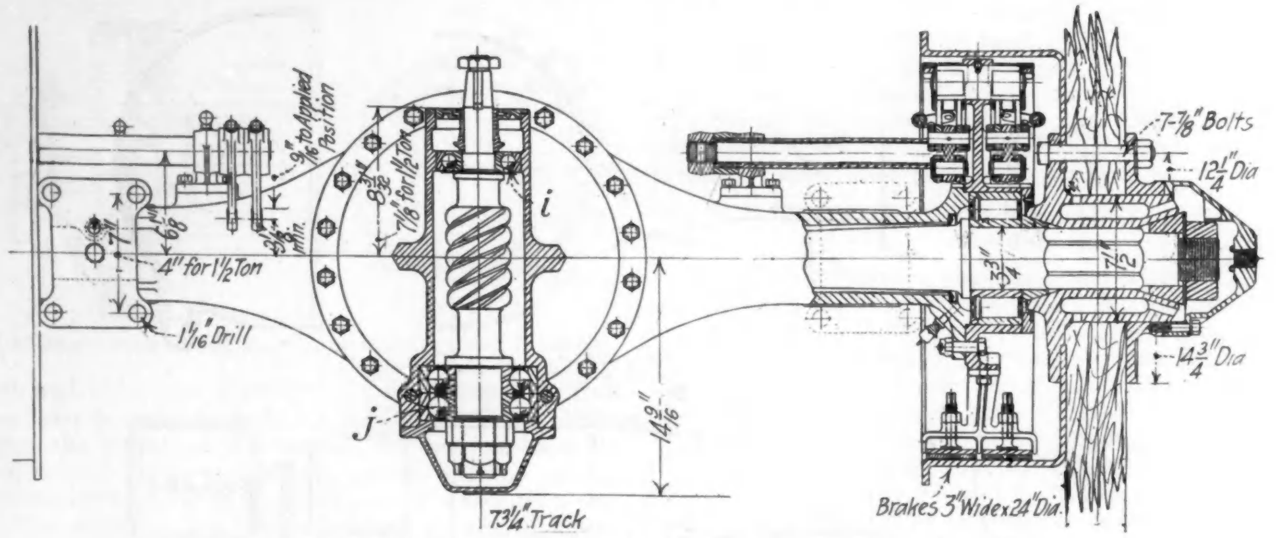


FIG. 9—REAR-AXLE SUITABLE FOR TRUCKS HAVING A CAPACITY OF FROM 5 TO 6 TONS

the bowl of the axle housing, and as the wormwheel rotates all the bearings and the worm and wormwheel are lubricated by the splash. The only other parts needing lubrication in the entire rear-axle are the brake rocker-shafts and the wheel bearings, and these are lubricated by oil-cups provided at these points.

The lower part of Fig. 11 is a detail of the axle shaft, made of S.A.E. No. 2340 heat-treated steel. The shafts are forged to size from small billets. The ends of the axle where it enters the differential, as well as where it drives the wheel, are hexagonal. The diameter at the wheel bearing-seat is $3\frac{3}{4}$ in. and the distance across the flats of the hexagonal end at the differential is 2 7/16 in. The method of attachment to the wheel can be seen from the assembly drawing in the upper portion. The wheel is seated by collets *k*, which are jambed by tightening the axle nut, and this in turn is locked by the hub-cap. The hub-cap is attached to the rear hub-flange by three $\frac{1}{2}$-in. studs and lock-washers. The two brakes are placed side by side in the drum, the same as in the Class B axle. In the Sheldon axles the cam type of brake is used on the smaller sizes, while on the larger sizes the wrap-up type is employed.

Fig. 12 shows the semi-floating worm-drive rear-axle of the Wisconsin Parts Co. The worm and worm-gear complete are mounted on the differential carrier and inserted into the housing from above and attached thereto. The worm shaft is mounted on ball bearings, the forward one being free to float; the outer race is therefore not confined endwise. The wheel is

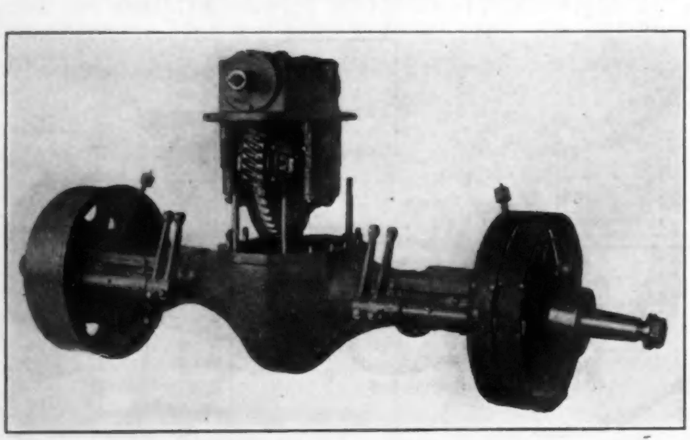


FIG. 10—WORMWHEEL AND WORMWHEEL CARRIER USED WITH THE AXLE ILLUSTRATED IN FIG. 9

mounted on double-row ball bearings or on straight roller-bearings. The wormgear is held between the differential flanges and is piloted, thus being supported throughout its circumference. To prevent egress of oil

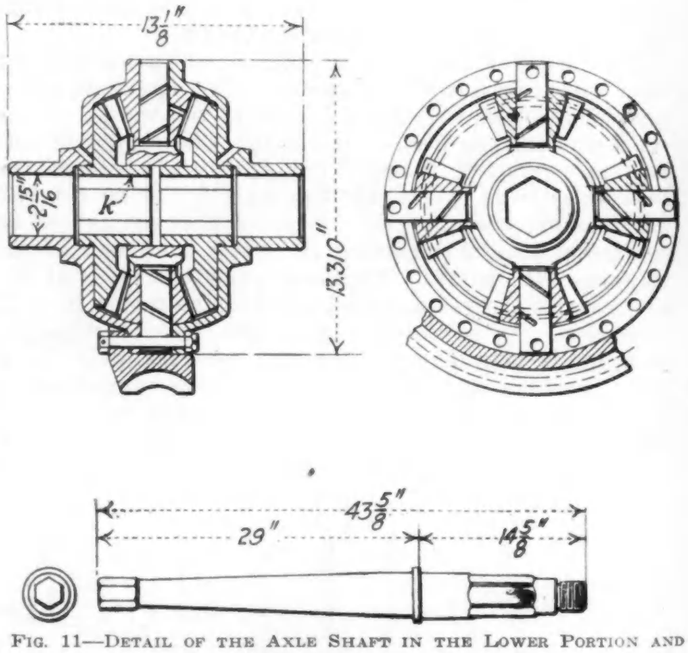


FIG. 11—DETAIL OF THE AXLE SHAFT IN THE LOWER PORTION AND ABOVE THE ASSEMBLY DRAWING SHOWING THE METHOD OF ATTACHING THE WHEEL TO THE AXLE

through the wheel bearing onto the brakes, two felt washers *l* are provided, one just behind the wheel bearing and one in the adjusting nut. The brakes, as may be noted, are both internal and mounted on a *double* brake drum. The drums are placed concentrically, thereby permitting the use of comparatively narrow brake-drums which enables the wheels to be located closer to the spring-seats. The housing is a malleable casting. The drive-shafts are made of No. 3140 S.A.E. chrome-nickel steel, heat-treated to show an ultimate strength of about 200,000 lb. per sq. in.

Fig. 13 shows the axle constructed by the Timken Detroit Axle Co. for trucks of from 2 to 5-ton capacity. In these models this company uses the full-floating axle, while in its smaller models, having capacities of from $\frac{3}{4}$ to $1\frac{1}{2}$ tons, it uses the semi-floating type. All the Timken axles are worm-driven, and the entire construc-

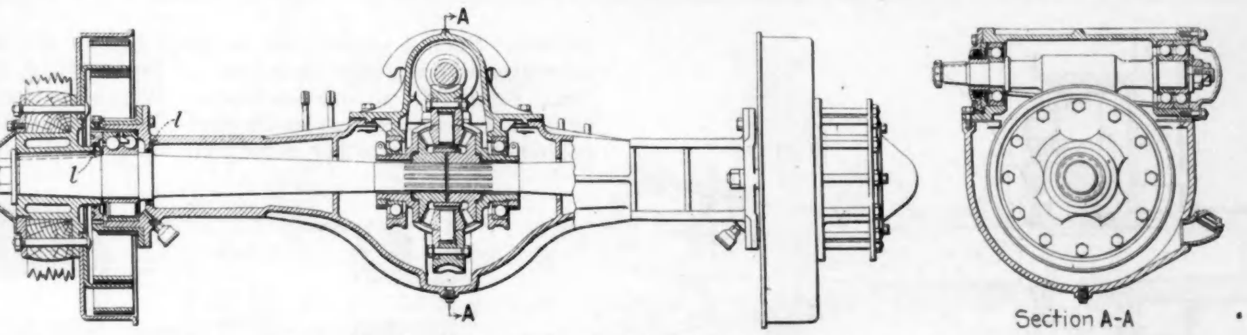


FIG. 12—A SEMI-FLOATING WORM-DRIVE REAR-AXLE

tion is similar to the Class B axle, the chief difference being in the brakes.

Fig. 14 is a side view of the brakeshoes in the brake-drum. There are four shoes, *m, n, o* and *p*, each extending about one-fourth of the circumference of the drum, but covering full width, as seen from the assembly at *q*. The brake cams, *r* and *s*, which are operated by the brake-shaft levers *t* and *u* respectively, actuate two opposite shoes; *s* actuating *m* and *n*, and *r* operating *o* and *p*.

Fig. 15 is the fixed-hub or semi-floating type of Timken rear-axle. This differs from the type illustrated in Fig. 13 in that the axle shaft is fixedly attached to the wheel direct, instead of by a splined driving-plate, and the axle supports the load of the wheels. In full-floating axles there is a greater stress on the axle housing, near the wheel, than in the fixed-hub type.

The Pierce-Arrow $3\frac{1}{2}$ and 5-ton truck axles are full-floating; the wheels are provided with taper roller-bearings, while the worm and wormwheel are supported on ball bearings. A propeller-shaft brake is used as the service brake, while the emergency brake acts on the inside of the brake-drums, as seen in Fig. 16, which is a cross-section of the rear-axle assembly. In its 2-ton model this firm employs taper roller-bearings throughout and also for the worm. The axle is semi-floating; the two brakes are placed side-by-side in the rear-wheel brake-drums. Instead of driving through torque and radius-rods, the Hotchkiss drive is employed in the small models.

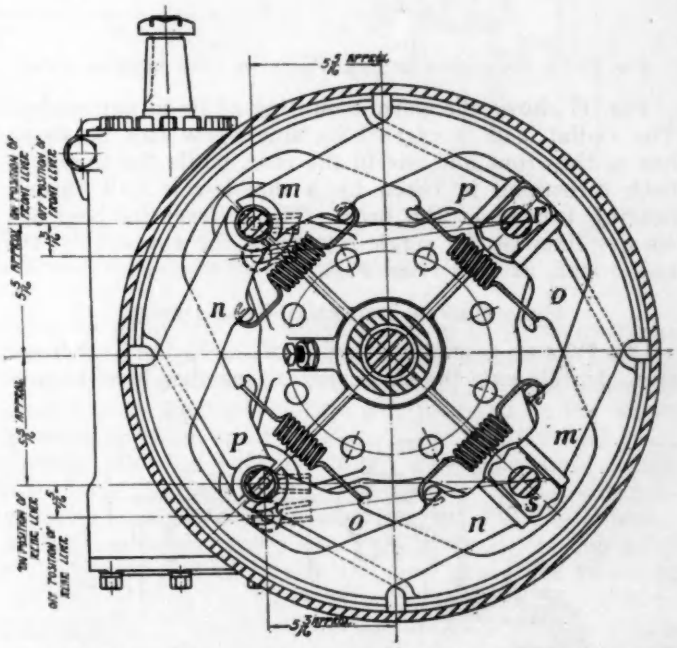


FIG. 14—A SIDE VIEW OF THE BRAKE-DRUM USED WITH AXLE SHOWN IN FIG. 13 ILLUSTRATING THE ARRANGEMENT OF THE BRAKE-SHOES

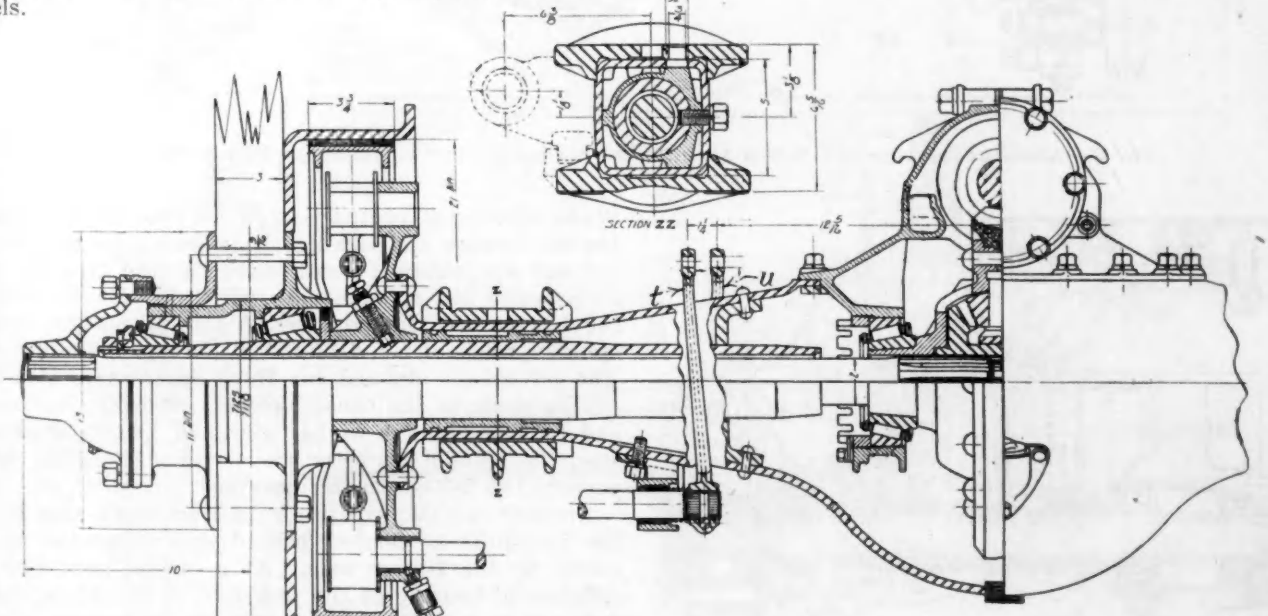


FIG. 13—A FULL-FLOATING AXLE FOR TRUCKS RANGING FROM 2 TO 5 TONS IN CAPACITY

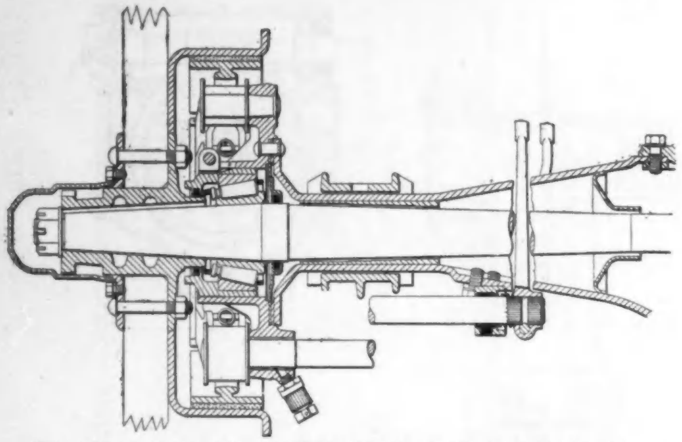


FIG. 15—A FIXED-HUB OR SEMI-FLOATING TYPE OF REAR AXLE

Fig. 17 shows the worm mounting of its larger models. The radial load is carried by single-row ball bearings, one in the front and one in the rear, while the thrust in both directions is taken by a double-row ball thrust-bearing located at the back. The differential bearings consist likewise of single-row balls for supporting the radial load, as seen from Fig. 16.

EXAMPLES OF INTERNAL-GEAR DRIVES

Fig. 18 is an assembly drawing of the ¾-ton Torbensen axle. In this axle the load-carrying member is of I-beam section. Such a section can be made lighter for a given strength and a given direction of the vertical stresses than round or square sections. The objection cited against I-beam rear-axles is that the stresses are imparted to it from other directions than purely vertical.

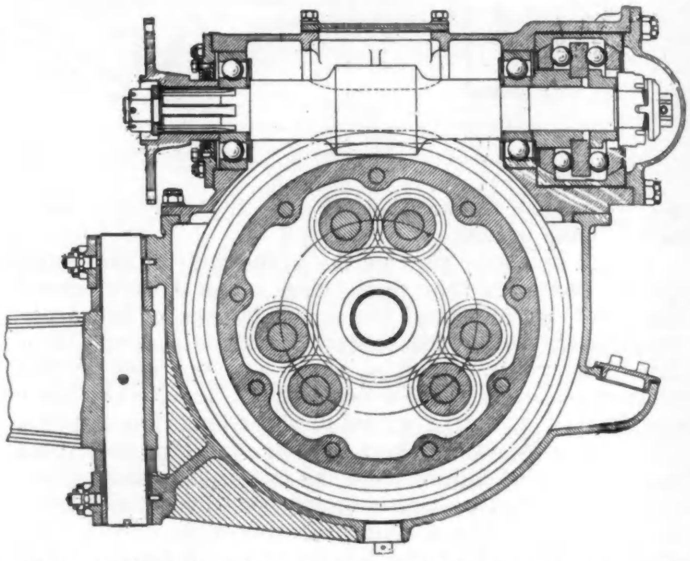


FIG. 17—WORM MOUNTING OF A LARGER TYPE OF AXLE SHOWN IN FIG. 16

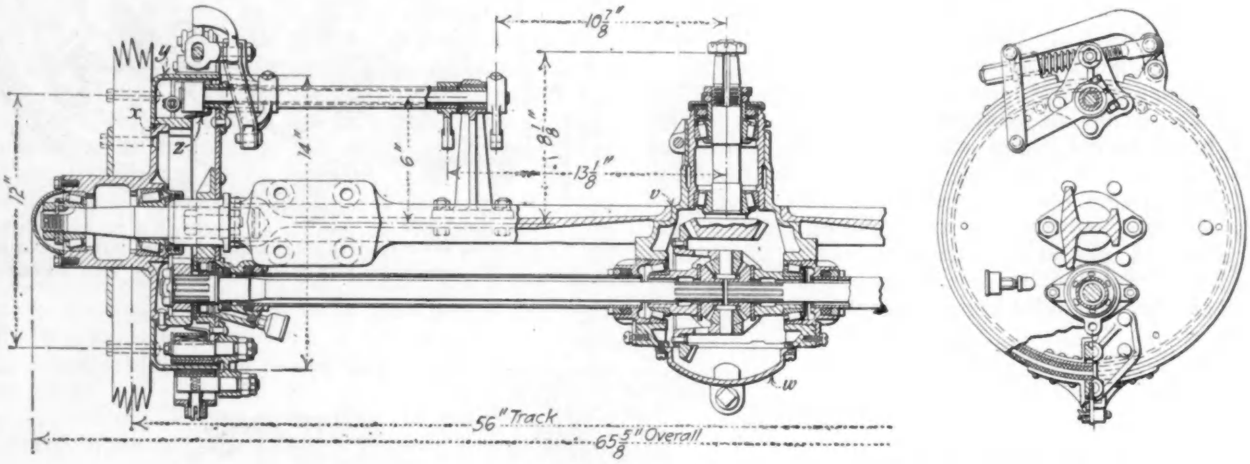


FIG. 18—ASSEMBLY DRAWING OF A ¾-TON AXLE HAVING THE LOAD-CARRYING MEMBER OF I-BEAM SECTION

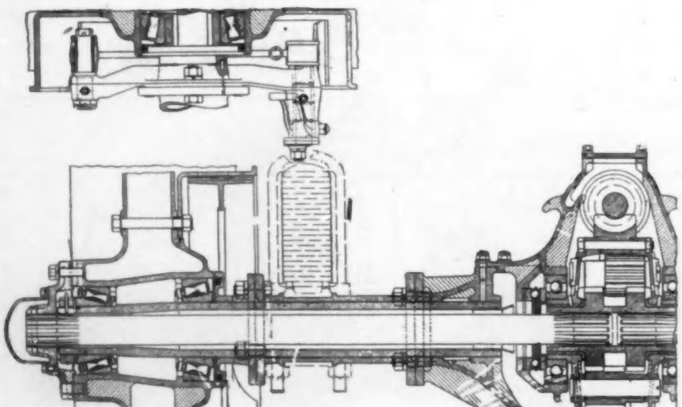


FIG. 16—CROSS-SECTION OF THE REAR-AXLE ASSEMBLY FOR TRUCKS RANGING FROM 3½ TO 5 TONS IN CAPACITY

When striking an obstruction on the road, or when applying the brakes and locking the wheels, the stresses are not entirely vertical; however, by far the greatest loads in practice are substantially vertical. At the ends of the I-beam are tapered holes, 4¼ in. deep, for inserted spindles, made of No. 6130 S.A.E. chrome-vanadium steel. The advantage claimed for these spindles is that they can be made to the exact hardness desired more easily, and by employing a higher grade of heat-treated alloy steel the spindle diameter may be made smaller, which permits the use of smaller bearings.

Another special feature of the Torbensen axle is that the jackshaft is carried behind and supported in the center by the I-beam axle. At v can be seen how the differential housing of the jackshaft is shouldered where it is attached to the I-beam axle by cap-screws. Locating the differential housing behind the load-carrying member permits the removal of the housing cover w for inspection and adjustment of the gears and bearings. The advantage claimed for the feature of attaching the

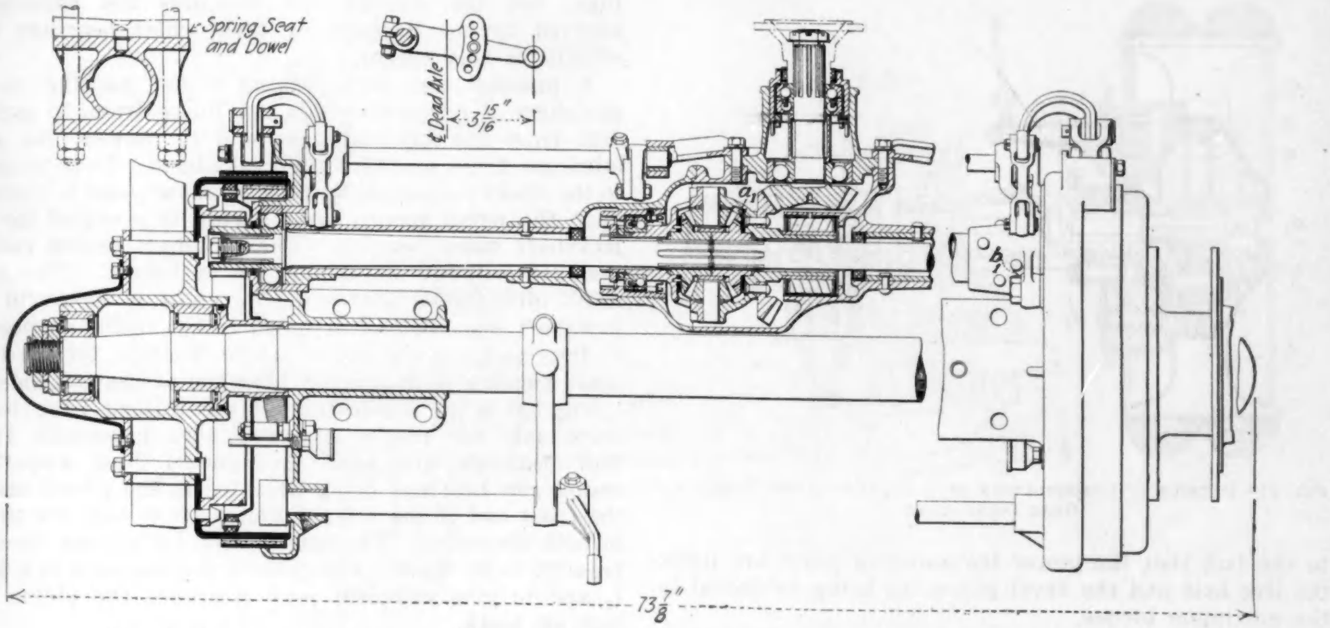


FIG. 19—AN EXAMPLE OF A ROUND INTERNAL-GEAR AXLE MADE FROM CHROME-NICKEL BAR STOCK

differential housing rigidly to the load-carrying member is that under any deflection of the axle the jackshaft will flex with it and thus maintain the line contact of the spur gears, whereas, if these members are not attached together rigidly, any strain in the load-carrying member will deflect it and, since the shaft with the pinions is not deflected, only the outer edges of the teeth will come in contact with the internal gears.

The internal gear is attached to the wheel hub by a press fit and 12 5/16-in. rivets, as shown at x in the assembly drawing. This is claimed to be a distinct advantage over the constructions where the ring-gear is held in place by bolts passing through the gear and the woodwheel, where the gear is seated on a wooden surface. Attaching the gear to the hub direct provides a metal-to-metal contact, which tends to keep the gears in line, while when it is attached to the wheels by bolts it is liable to distortion when drawing up the bolts in assembling. The brake-drum y is mounted on the wheel to admit of some space between it and the internal gear, thereby allowing sufficient room for the brakes, which may both be placed inside the drum, if the latter is widened. In the drawing one internal and one external brake are shown, and a sheet metal ring z is employed to keep dust and dirt from the spur gears and to shield the brakes from the lubricant of the gears. The attachment of the brake-shaft bracket to the I-beam is shown in the assembly. It is stated by the makers that this ¾-ton axle weighs about the same as a bevel-gear-drive axle suitable for a 3000-lb. passenger car, due chiefly

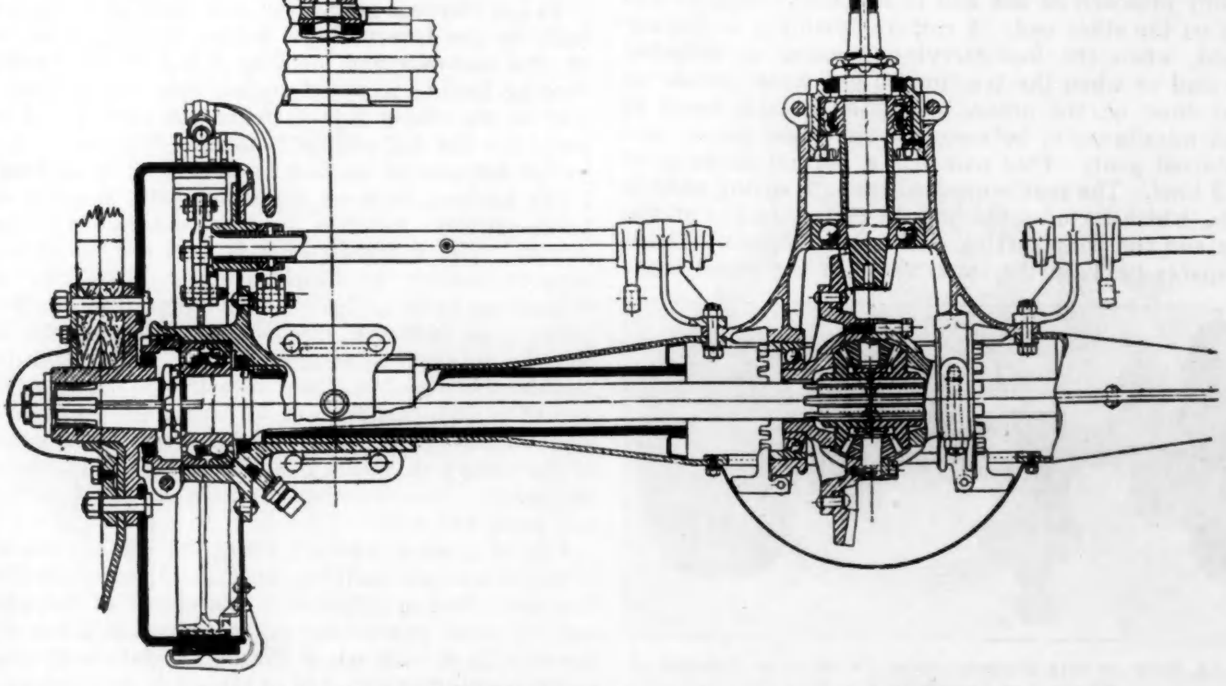


FIG. 20—A 1-TON BEVEL-DRIVE AXLE FOR TRUCKS EQUIPPED WITH PNEUMATIC TIRES

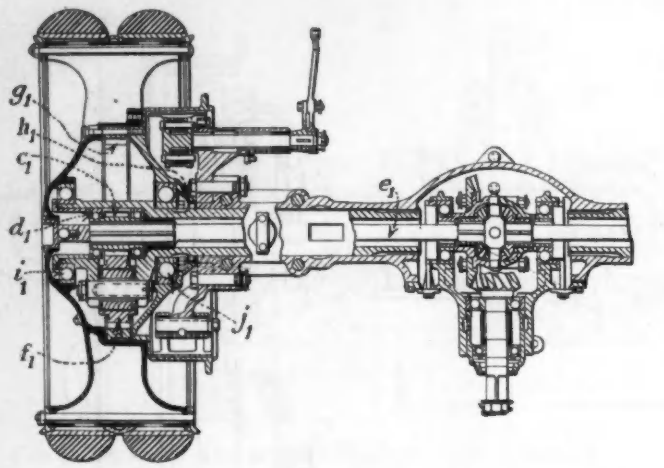


FIG. 21—DETAILS OF CONSTRUCTION OF A 3½ AND 5-TON INTERNAL-GEAR DRIVE AXLE

to the fact that the power transmitting parts are light, the live axle and the bevel gears not being subjected to the maximum torque.

In the Russell internal-gear axle, shown in Fig. 19, a round axle, made from chrome-nickel bar stock, is used. It is heat-treated at the mill, which is claimed to give uniform physical properties throughout. The load-carrying member is behind the jackshaft, which permits the use of a shorter propeller-shaft. Furthermore, by having the jackshaft pinion in front of the wheel center the pressure of the driving pinions on the internal gear is downward, and this reduces the load on the wheel bearings considerably. If the pinions are behind the dead axle, the bearing pressures arising from the drive are added to those due to the static load on the wheel.

The jackshaft housing is entirely independent of the load-carrying axle, and it is not rigidly bolted to it. With all load carriers there is some deflection due to the strain in the metal, especially under severe service, and it is claimed that by not having the jackshaft rigidly attached to the axle no part of the strain of the load-carrying axle is transmitted to the jackshaft. The jackshaft housing is rigidly attached at one end to the brake support and loosely on the other end. A certain flexibility is thereby obtained, when the load-carrying member is deflected under load or when the traction on one wheel should be greater than on the other, which, it is said, tends to prevent misalignment between the spur-gear pinion and the internal gear. This axle has a normal capacity of about 2 tons. The maximum load on both spring-pads is 8100 lb., which includes the portion of the weight of the chassis and the body resting on the spring-pads. There is no spacer between the inner races of the wheel bearings, but the construction embodies the adjustment adopted by the Society for front-wheel bearings that eliminates this spacer.

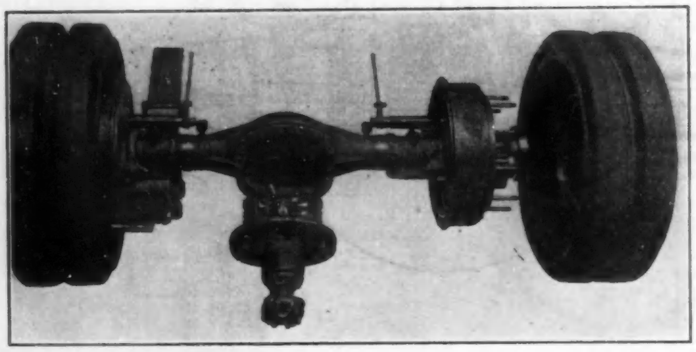


FIG. 22—A VIEW OF THE INTERNAL-GEAR DRIVE AXLE, DETAILS OF WHICH ARE SHOWN IN FIG. 21

A pressed-steel cover having a felt packing on its periphery is mounted within the brake-drum, to exclude dirt from the internal gear and to prevent the gear lubricant from reaching the brake-band. It is attached to the brake support a_1 by bolts as shown, and is centered upon the wheel spacer. A felt ring is provided for the jackshaft outer bearing. The jackshaft pinion can be removed without disturbing the jackshaft. The jackshaft torque-yoke shown at the right is keyed to the jackshaft housing and attached to the brake support at b_1, thus carrying the entire torque reaction, for the jackshaft housing is not rigidly attached at the left side.

Fig. 20 is an illustration of the Russell 1-ton bevel-drive axle for trucks equipped with pneumatic tires. Ball bearings are used throughout, New Departure double-row bearings being provided in the wheels and at the front end of the drive-pinion shaft to take the thrust in both directions. The axle is built for a truck-speed of from 25 to 35 m.p.h. The ratio in the rear axle is 6.33 to 1, and to give sufficient road clearance the pinion has only six teeth.

Fig. 21 shows the construction of the 3½ and 5-ton internal-gear drive axles of the White trucks illustrated in Fig. 22. In these axles the White Company uses one internal brake at the rear wheels, serving as the emergency brake, and a foot-operated propeller-shaft service brake. The first reduction is obtained by spiral bevel-gears in the center of the axle and the second by the pinion c_1, Fig. 21, which is supported by two ball bearings d_1 and splined to the axle or drive-shaft e_1; this pinion meshes with an idler or intermediate gear f_1, running on Hyatt roller bearings, which in turn meshes with the wheel ring-gear g_1, which is plainly visible at the right of Fig. 22. The joint between the wheel and axle housing is near the housing center at h_1, Fig. 21; hence the gears can run in a bath of oil with less danger of leakage than if there were a joint at, or near, the periphery of the drum. The wheels are full-floating, each wheel being supported by two ball bearings, i_1 and j_1, placed a great distance apart to reduce the bearing pressures.

In the internal-gear-drive axle used on the 1-ton truck built by the International Harvester Co., shown in Fig. 23, the load-carrying member k_1 is a chrome-nickel steel forging having a round section like the Russell, except that in the center it is flattened out and curved to make room for the differential housing. The latter is piloted to the differential carrier as shown. The driving-shafts l_1 are exposed between the differential housing and the brake-support castings which contain the spur-gear pinion. The internal gear is enclosed by this brake-support casting to which are fitted the two internal expanding brakes. All the bearings are of the Hyatt roller type with the exception of a double-row bearing for the internal spur-gear pinion. The lateral thrust against the wheel in either direction is taken by the two sets of thrust-washers, one of bronze placed between two steel washers. All the Hyatt bearings in these axles are of the same size, which is an advantage when repairs are necessary. The clearances and the drive fits of the various parts are noted.

Fig. 24 gives a general view of the 1 to 1½-ton Walker-Weiss truck axle built by the Flint Motor Axle Co. The first reduction is obtained in the center of the axle by a pair of bevel gears; the other reduction takes place at the outside of each wheel through a stationary ring gear and three pinions attached to the wheel driving-cap. This

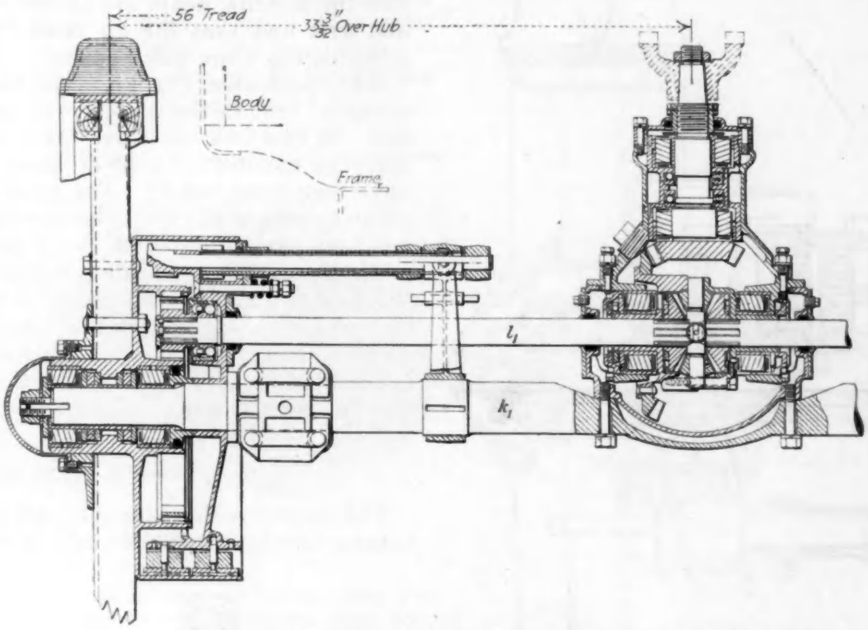


FIG. 23—INTERNAL-GEAR-DRIVE AXLE USED ON A 1-TON TRUCK

is said to reduce the size of the gears required. By having a plurality of pinions the tooth pressure is small as compared with the single pinion drive; hence smaller gears may be employed. The main shaft, made of chrome-nickel steel, has attached to its outer end a gear pinion which meshes with three planetary or idler gears that are in mesh with the stationary ring-gear. When the shaft rotates the pinion m_1 at its outer end drives the idler gears n_1. The ring-gear v_1, being stationary, takes the reaction from the idler gears. In this manner these idler gears, which are studded to the wheel driving-flange p_1, rotate about the studs and, since the ring-gear is stationary, the driving-flange will be rotated in a forward direction, the same as the drive-shaft. The housing, which covers the driving-shaft, serves as the load-carrying member. The wheel bearings, the driving pinions and idler pinions run in a bath of oil. As in full-floating axles in general, the gears can be replaced without jacking-up the truck or removing the wheels. It is claimed that in this construc-

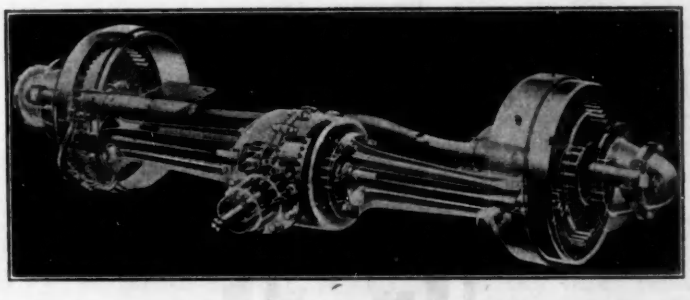


FIG. 25—AN AXLE WITH A ROUND LOAD-CARRYING MEMBER THAT IS BENT IN THE CENTER TO CLEAR THE DIFFERENTIAL HOUSING

FIG. 24—A 1 TO 1½-TON DOUBLE-REDUCTION TRUCK AXLE

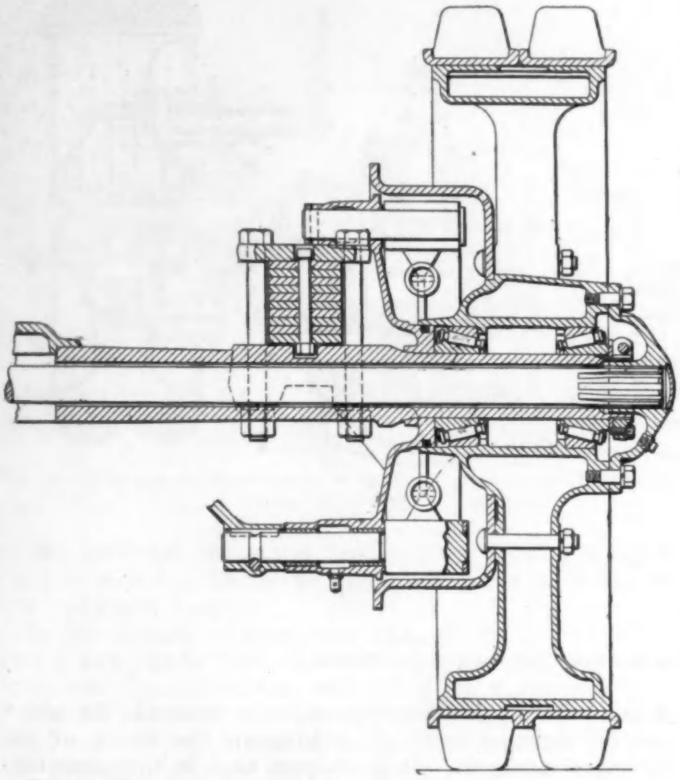


FIG. 26—CROSS-SECTION THROUGH THE REAR WHEEL OF DOUBLE-REDUCTION AXLE

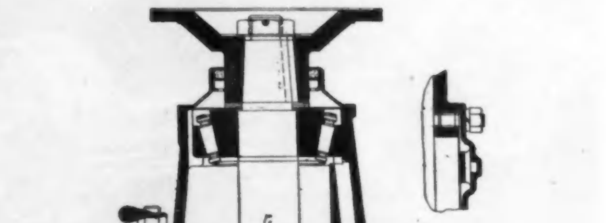


FIG. 27—SECTION THROUGH THE DOUBLE REDUCTION OF THE SAME AXLE AS ILLUSTRATED IN FIG. 26

tion the internal gears are protected absolutely from dust and grit and that the oil flows from one wheel to the other through the axle housing and the differential.

The Clark axle, Fig. 25, employs a round load-carrying member, bent in the center to clear the differential housing. In this axle the drive-shaft is in front of the load-carrying member; the advantages of such a construction have been cited before. The brake-drum contains a partition to separate it from the internal gears. Hyatt roller-bearings are employed to carry most of the radial load, while the double-row ball-bearings at the outer ends of the axle and at the pinion shaft are used to carry a portion of the radial load and to take the thrust in both directions, and thrust ball-bearings are installed behind the differential housing. The internal gear is attached to the hub by bolts passing through the gear, the brake-drum and the wheel spokes.

THE DOUBLE-REDUCTION AXLE

The term double-reduction axle as employed today denotes a two-fold reduction in the center of the rear-axle

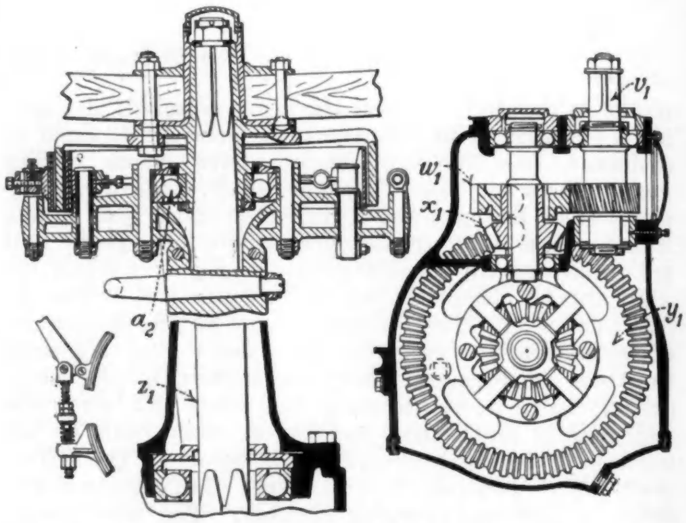


FIG. 28—ASSEMBLY OF A 1½ TO 2-TON DOUBLE-REDUCTION TRUCK AXLE

housing. In some of the double-reduction axles the first reduction is accomplished by spur gears or spiral spur gears, as in the White 1½ to 2-ton axle, and the second reduction by bevel gears; while in others, like the Autocar and the small Mack truck, the first reduction is by bevel gears and the second by spur gears. The International Motor Co., which used a worm, as well as a chain, drive on its 1½ to 2-ton model for several years, has recently abandoned the worm drive altogether and substituted a double-reduction axle.

The main axle-chamber is a drop-forged banjo or yoke, with hollow tubular ends carrying the axle shafts, as in the conventional live axle, with the driving members enclosed within the banjo yoke. The drive from the propeller-shaft is in a straight line, passing first through a pair of bevel and then through a pair of spur gears to the differential. The banjo is inclined at an angle of 45 deg., to be in a better position to resist road shocks, when hitting obstructions on the road, which are normally oblique in direction.

Fig. 26 shows a cross-section through the rear wheel, while Fig. 27 shows a section through the double reduction, including the differential; the flange q_1 being a part of the housing containing the bevel and spur gears. The pinion is made integral with the rear-axle

pinion shaft r_1; the latter drives the bevel gear keyed to jackshaft s_1, which is integral with spur gear t_1. This spur gear meshes with the big spur or bull gear u_1. All bearings in this axle are of the Timken taper-roller type; the brake-shafts have self-lubricating bushings. The advantages claimed for the double-reduction axle are that its efficiency is constant throughout

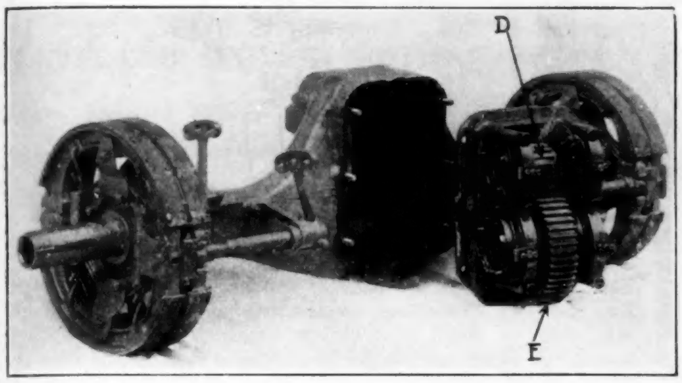


FIG. 29—A DOUBLE-REDUCTION REAR-AXLE IN WHICH THE FIRST REDUCTION IS BY BEVEL GEARS AND THE SECOND BY SPUR GEARS

speed and load ranges and wear and that it is more durable than the worm and the internal-gear drives and, like the worm drive, free from dust.

Another example of a double-reduction model is the White 1½ to 2-ton truck axle, an assembly of which is shown in Fig. 28. The first reduction is obtained by spiral spur-gears, the drive coming through the pinion countershaft v_1, with which the spur gear is made integral, the latter meshing with spiral spur gear w_1. This gear is keyed to the hub of the bevel pinion x_1, which meshes with the bevel driving gear y_1. The axle or drive shaft z_1 is semi-floating, the wheel being supported by ball bearing a_2.

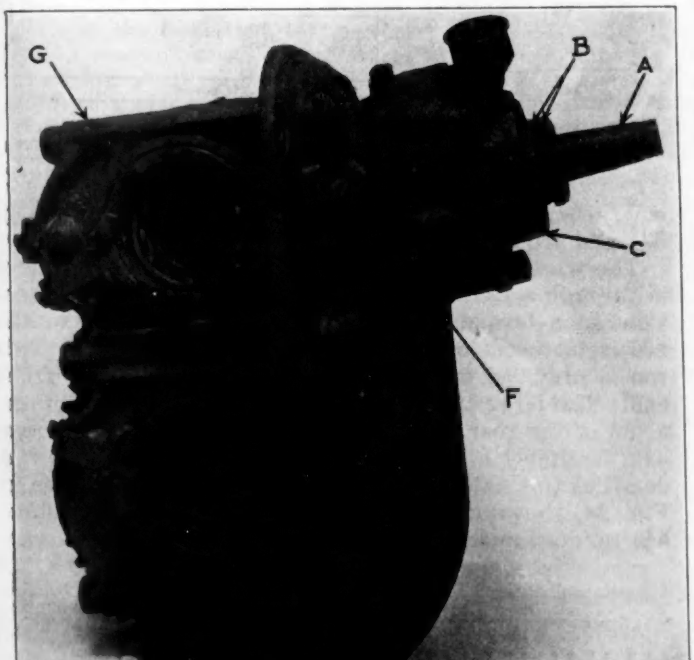


FIG. 30—A DOUBLE-REDUCTION REAR-AXLE IN WHICH ALL THE GEARS ARE ASSEMBLED ON THE FRONT COVER

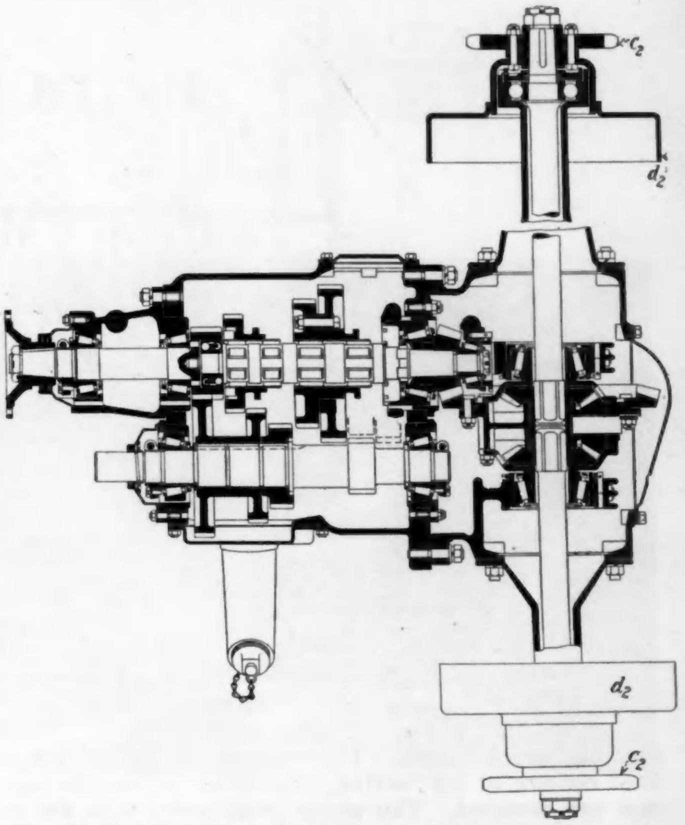


FIG. 31—SECTIONAL VIEW THROUGH THE TRANSMISSION AND JACK SHAFT OF A CHAIN-DRIVEN TRUCK

Figs. 29 and 30 show the double-reduction rear axle used on the Autocar truck. In this axle the first reduction is by bevel gears, and the second reduction by spur gears. All the gears are assembled on the front cover, shown in Fig. 30, while, for inspection and adjustment, a cover-plate can be removed from the back of the axle housing.

The pinion shaft *A* is mounted on two taper roller-bearings, which can be adjusted by the two nuts *B*. The pinion shaft and bearings are all mounted in a separate case *C* attached to the front cover. The front end of the pinion shaft is splined. A bevel pinion is pressed on this

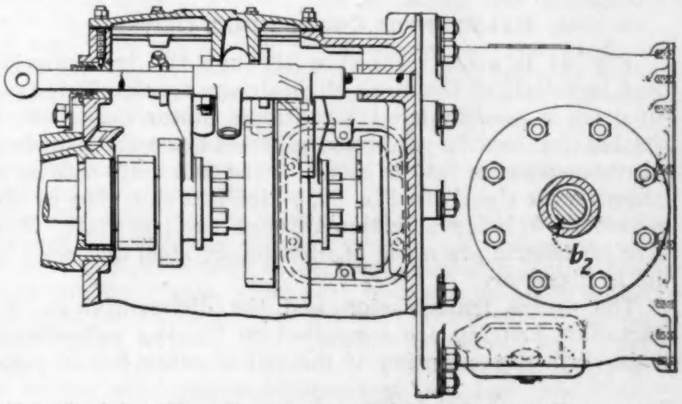


FIG. 32—A SIDE VIEW OF THE ASSEMBLY OF THE CHAIN-DRIVEN AXLE ILLUSTRATED IN FIG. 31

and is held in place by a nut and washer. This bevel pinion meshes with the bevel gear on the jackshaft which forms a unit with the small spur gear *D*, Fig. 29, which in turn meshes with the differential spur ring-gear *E*.

By removing plug *F*, Fig. 30, the adjustment of the bevel gears, when the cover assembly is in the axle hous-

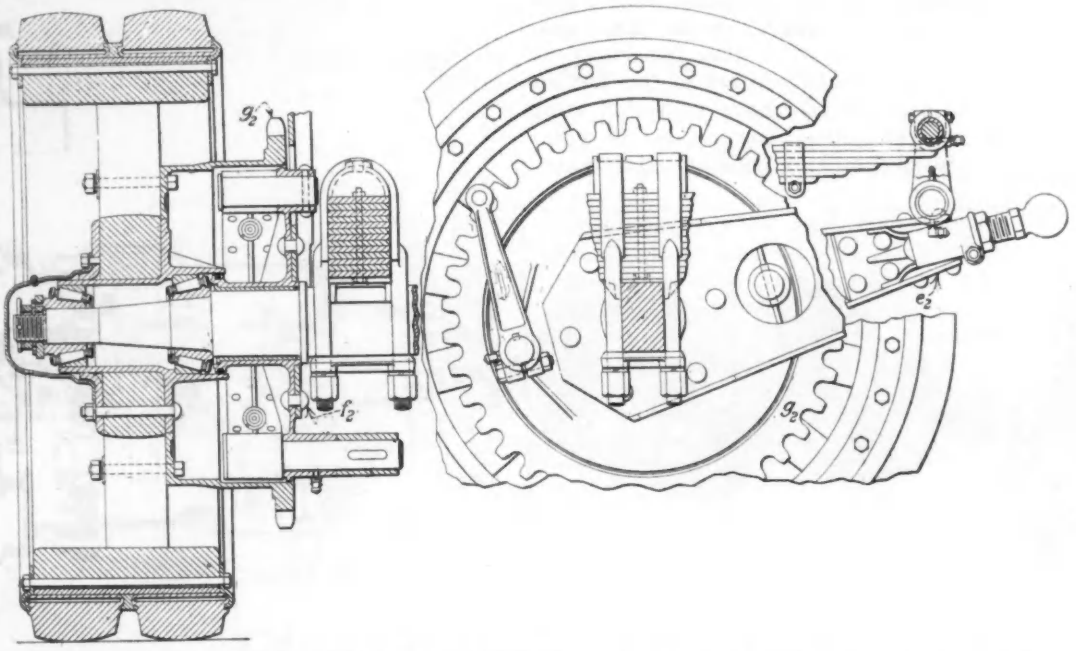


FIG. 33—CROSS-SECTION AND SIDE ELEVATION SHOWING THE LOCATION OF THE EMERGENCY BRAKES IN THE REAR WHEELS

ing, can be examined. The spur gears are located on fixed centers of the casting, into which adjustable bearings are mounted. The whole front cover, with the exception of the bridge *G* on the pinion shaft case *C*, is a single casting; this bridge is used as a clamp to hold the bearing adjusters and retainers in position. By employing spur gears between the jackshaft and the differential ring-gear there is no side-thrust against the ring-gear. The axle is of the full-floating type. The wheel is mounted upon the tube that carries the wheel bearings. This tube is pressed into the axle housing under heavy pressure and then held in place by set-screws. In the event of any accident, this tube will sustain the damage rather than the axle housing and can be replaced at comparatively low cost. In the heavy-duty trucks the axle casings are of square section, and the tubes are pressed all the way through the casing into a web, close to the differential gears, thus reinforcing the casing.

EXAMPLES OF CHAIN FINAL DRIVES

Fig. 31 is a sectional view through the transmission and jackshaft of the Mack chain-driven truck. This type of drive is used by the International Motor Co. on all of its heavier models. As may be noted from the drawing, the transmission case is bolted to the jackshaft case as is shown more clearly in Fig. 32, which is a side view of the assembly, b_2 being a section through the jackshaft. The live jackshafts are made of nickel-alloy steel and are 1¾ in. in diameter.

The entire transmission and the differential in the jackshaft housing are supported on Timken roller-bearings, with the exception of the self-aligning S.K.F. pilot bearing in the pinion gear and the ball bearings at the outer ends of the shafts. The service brake-drum hubs are keyed to the outer tapered ends of the shafts and the driving sprockets c_2, Fig. 31, are bolted to these hubs. The service brakes on the drum d_2 are of the external contracting type, 5 in. in diameter and 3 in. wide. The

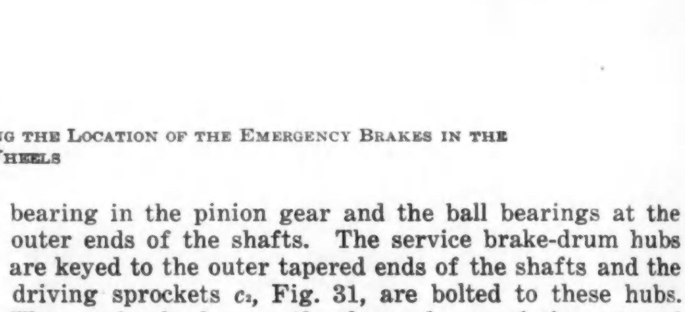


FIG. 34—DETAIL OF THE RECTANGULAR REAR-AXLE OF A 5-TON CHAIN-DRIVEN TRUCK

emergency brakes are located in the rear wheels (see Fig. 33) and are 20 in. in diameter and 3½ in. wide.

The wheels are of the full-floating type and the drive is through a radius-rod e_2 which also performs the functions of a torque and thrust-rod. The rear end of the rod is fastened to the brake supports by rivets f_2. This rod is provided with a ball-and-socket joint at its front end. The large sprockets g_2 are integral with the brake-drum of the rear wheels. In chain-driven axles the rear axle is simply a load-carrying member or dead axle. A detail of this axle as used on the 5-ton model is given in Fig. 34, showing this axle to be of rectangular section, 4⅜ in. deep and 2¾ in. wide.

WINTER TESTS SHOW LOWER MILEAGE WITH HEAVY FUELS

IN the paper with the above title that was presented at the recent Semi-Annual Meeting by Dr. H. C. Dickinson and John A. C. Warner and that was printed in the July issue of THE JOURNAL a typographical error was made in giving the increase in the average fuel-consumption in the fourth line of the second column on p. 91. As printed the statement was made that "*Under winter conditions* there is a small but definite increase in average fuel-consumption of about 30 per cent." This figure should be 3 per cent, as would be perfectly obvious from the context.

SOCIETY MEETINGS

CLEVELAND SECTION VISITS CANTON

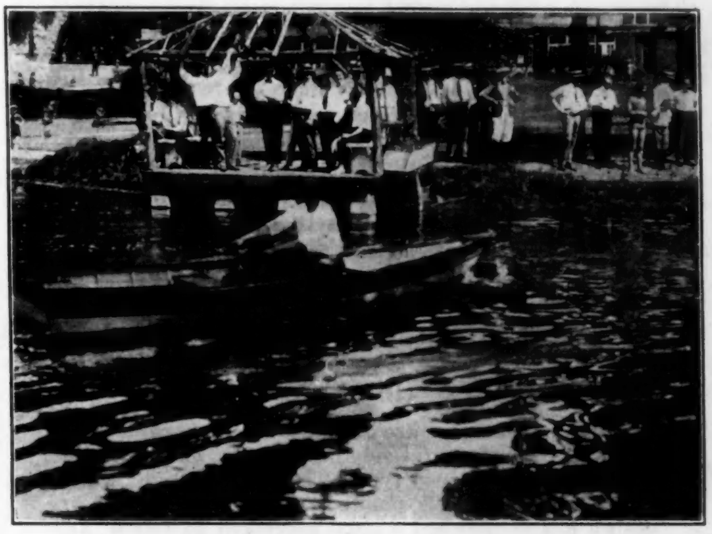
Members Inspect Roller Bearing Factory and Enjoy Afternoon of Sports

It was a heyday in Canton on July 13. Despite the age-worn prejudice set up against Friday the 13th by those addicted to superstition, some 65 members of the Cleveland Section of the Society gathered in the village made famous by tapered rollers to disport themselves as the guests of H. H. Timken and his diminutive supporters, Porter and Buckwalter. John Younger writes that "only a schoolgirl with a rich choice and selection of adjectives could do justice to the party," and we have no reason to question the veracity of Cleveland's energetic chairman.

The morning was spent in making a very thorough inspection of the Timken Roller Bearing Co.'s plant, the party being escorted by guides who had been schooled to explain manufacturing operations to the minutest detail. The Timken factory is unique in that it houses under one roof all of the operations necessary to the production of anti-friction bearings from nature's basic materials. The members were enabled to see the production of molten steel, the pouring of the ingots, and the rolling of the blooms. The manufacture of tubing, bar-stock and wire proved to be intensely interesting. Every step in the fabrication of finished bearings was explained to the visitors. The inspection was concluded with a visit to the departments where the hardening, the grinding and the final assembly were carried on. All agreed that the Timken factory represented the last word in efficiency and presented a highly interesting show place because of the great diversity and comprehensiveness of the operations carried on within its walls.

Following the inspection, the members partook of a Timken lunch and adjourned to the shores of Canton's famous Congress Lake. Here the party split into three sections; the followers of Gene Sarazen's pet diversion hied themselves to number one tee; students of Izaak Walton rippled the placid waters with hook and sinker, and the wakeful ones of the remaining assembly disported themselves in heated contests of brain and brawn. Prizes were given to the winners of the numerous events, and the struggle for these embittered the combatants to a state of teeth-gnashing. The tug-of-war setting was devised by some engineer with a creative mind and a respect for action. The rope was stretched across a narrow creek whose banks were of about the same consistency as 600 W. The opposing teams battled to see which one would have the privilege of negotiating the ford, truly an enticing reward. The canoe-tilting event reflected exceptionally intimate knowledge of modern naval tactics on the part of the participants. Dr. Georg Madelung, bird of the air, proved to be accomplished in submarine conduct also; the Armstrong brothers, in deference to their name, proved the value of teamwork in the tilting event.

The prize fish of the day was captured by W. H. Lolley after a hard tussle. Proud of its single ounce of weight, this diminutive representative of Canton's finnish populace



TUNING UP FOR THE BOAT RACES ON CONGRESS LAKE

met its master and succumbed. Stuart Cowan sank the winning putt on the eighteenth green and Ernest Wooler sported the healthiest drive in the golf events. B. H. Blair drew the violet ribbon for his all-round efficiency in the athletic contests. His victory in the shot-put was made possible by the barring of "Tiny" Buckwalter because of the latter's affiliation with the Timken clan, which was acting as host of the day.

The completion of the sports events found the majority of the contestants completely deenergized and possessed of the proverbial appetite of the small boy. Though less active in body, the spectators likewise had become exhausted from the violent exertions demanded by boisterous mirth during the afternoon's carnival. The three units of the party were reassembled at eventide and appeased their respective hungers at the Timken board. The gathering was addressed briefly by H. H. Timken, who had proved to be himself an able host. He related how years ago he had journeyed to New York City to hire an engineer and had returned with Herbert W. Alden, president of the Society, to whom he was forced to pay more than he, Timken himself, was getting. The prizes were awarded to the victors of the day, songs were sung and a weary but happy group of engineers returned to their respective havens.

The Cleveland Section is carrying its active program of meetings through the summer season, specializing in instructive inspection trips through representative automotive manufacturing plants. The next visit will be made to the factory of the Firestone Tire & Rubber Co. in Akron during the month of August. Full particulars of the meeting arrangements can be secured from L. L. Williams, secretary of the Section, who may be addressed at 1051 Lakeview Road, Cleveland.

Tendencies in High-Speed Marine-Engine Design

By J. G. VINCENT[1]

MOTORBOAT MEETING PAPER

FOLLOWING the presentation of this paper by L. M. Woolson a general discussion was had by those present. In every case an effort has been made to have the speakers go over their remarks before publication in accordance with the customary practice of the Society. For the convenience of the members a brief abstract of the paper precedes the discussion with a reference to the issue of THE JOURNAL in which the paper appeared so that members who desire to refer to the complete text as originally printed and the illustrations that appeared in connection therewith can do so with a minimum of effort.

ABSTRACT

IN this paper there are discussed the general requirements of a comparatively new type of marine engine, that is, the high-speed type. The need for light weight is discussed and it is pointed out that aircraft-engine design has established a good precedent to follow, particularly with reference to crankshaft, connecting-rod, piston, cylinder and valve design.

Detailed discussion then follows concerning the requirements of this type of engine in respect to cooling, lubrication, starting, idling and general operation. The equipment to be furnished with the engine is covered in a general way and the installation requirements are touched upon.—[Printed in the March issue of THE JOURNAL.]

THE DISCUSSION

LEONARD OCHTMAN, JR.:—In any consideration of aircraft engines as applied to marine practice, we should take into account the fact that there are three supreme requirements for an aircraft engine; light weight, high power and minimum fuel-consumption. Those are desirable characteristics for a marine engine, but they cannot always be obtained because light-weight construction usually is at the expense of great reliability; also, it is very expensive on account of the extra machine work that must be done.

L. M. WOOLSON:—Reliability is a very pertinent factor that should have been covered. When first designed, the Liberty engine was required to go through a 50-hr. test, a test considered extremely severe; very few aircraft engines would go through a 50-hr. test at that time. We have progressed and the Navy is to be commended for its stand in taking-on all its aircraft engines only after a 300-hr. test. I believe the present automobile-engine record-test is 500 hr. Hence the aircraft engine is as reliable as the automobile engine. We tested an engine for the Navy and, at the end of 300 hr. that engine was practically as good as new. It developed something like 250 b. hp. during that test. It was dismantled and reassembled and the only work done was in regrinding the valves. How many strictly marine engines would go through that test without trouble, and how many would do anything like as well as this so-called light-weight aircraft-engine?

[1] M.S.A.E.—Vice-President in charge of engineering, Packard Motor Car Co., Detroit.

I take issue with the thought that, in a marine engine, the lack of weight means a lack of reliability. In some late Navy work, it has been shown that there is no connection between light weight and reliability. We are now laying down an engine some 200 lb. lighter than the Liberty engine, but it develops more power, is far more compact and has far lower stresses in all the main parts such as the connecting-rods, the bearings and all the parts that count. It is just a matter of studying where the weight should be put. The cylinder on this engine is of $5\frac{1}{8}$-in. bore; the weight of the cylinder is 9 lb. The only way we are able to get these high mean-effective pressures is by keeping the exhaust-valves very well cooled. I think everybody admits that it is practically impossible to cool an exhaust-valve that is seated on a thick cast-iron wall as well as one seated on a light steel-wall. It has been recognized that we cannot get this super performance and light weight at anything like the same expense that we can build the heavier engine; on the other hand, we must also recognize that the high-speed express-cruiser, for which these engines are intended, is essentially a rich man's hobby and must always be so.

MR. OCHTMAN:—It undoubtedly is true that aircraft-engine design has gone far ahead of that of any other type of engine within the past few years, and much can be learned from it as regards engine performance, such as power output and economy, and its reliability while retaining the light weight.

CHRIS SMITH:—I am an aviation-engine man and believe that there is much difference between a good aviation engine and a good marine engine. I am satisfied that the light, properly constructed aviation-engine will outwear any marine engine we have ever made. The marine engine man has not had the money with which to experiment. A moderate sized factory could produce all the marine engines made; consequently, the income from them has been limited.

The difference between the cast-iron and the steel connecting-rod lies in the weight, and yet the light rod can stand the gaff. If I were searching for reliability, I would choose the aviation engine. The engine is one part of the proposition and the boat is the other; the model of the boat must be very accurate. We used four Liberty engines in Miss America. She showed a little weakness along the forward keel the day before the races, and two 6-ft. lengths of sheet hard brass, 4 in. wide and 3/16 in. thick, were screwed to the bottom of the boat to stiffen the keel, where the stresses were coming. They chamfered the forward end of the brass and carried it back to the step. They took the boat out and could not keep her in the water. Afterward, they let the brass extend 3/16 in. into the wood to eliminate that rough edge and it worked all right. Much depends on the construction in getting a boat to perform just right.

W. C. WARE:—Engines that run more than 1400 r.p.m. are hardly in our class. Some of the marine-engine builders seem to think they have found a necessity for oil cooling even at that speed but, after conducting many

experiments we find we can do without it very nicely. We keep our oil-temperature down around 115 deg. fahr. without any difficulty. No trouble from very poor ventilation in engine compartments has been encountered. We have found a 30-deg. fahr. lower oil-temperature during the boat test than on the regular block-test which proves pretty well that good ventilation can be obtained in engine compartments. In reference to reverse gears, we are strongly of the belief that manufacturers should design reversing gears that can be flanged on to the rear end of the crankshaft. This does not mean much to manufacturers because they can put the present type of gear on without difficulty, but the man who has to take a reverse gear off "out in the woods" without proper gear pullers has a serious job on his hands.

CHAIRMAN L. C. HILL:—The matter of accessibility in marine-engine design is very important. In looking over some of the designs in the motorboat show, I find numerous cases where accessories such as starting and lighting generators cannot be removed without taking off many other parts on the boat or the engine. A few of the more progressive builders are adopting removable cylinder-heads. The average marine engine in the cock-pit of a boat is in about the most inaccessible position possible; it is certainly much easier to clean, grind valves and the like with the removable head and that should have been recognized in the motorboat field long ago. Accessibility may not be so important in boats using high-speed aviation-engines, because those people have plenty of mechanics and they usually lift the engine out of the boat to make major repairs. I have noticed boats with the oil coolers attached to the bottom of the crankcase. That necessitates the complete removal of the engine from the boat if there is ever any trouble due to any clogging up of the oil-cooling apparatus. Another example I have seen repeatedly is the installation of an engine with accessories mounted over the lag-screw holes in the crankcase flanges. That must be remedied. It is often impossible to get a lag screw of proper length into the space that is allowed between the bottom of the cylindrical generator and the hole in the crankcase flange.

H. L. BROWNBACK:—Many engines have oil gages in very inaccessible places so that, when a man fills the engine with oil, he cannot tell how much oil he has. In one case we have drilled through one of the studs going from the cylinder-heads down through the main-bearing caps to the sump and put a float and a rod with a little red ball on top so the man can see what the oil level is.

GEORGE W. WESLEY:—The reverse-gear manufacturers are absolutely confined by a cast-iron fence. They have a compartment on the rear end of the engine. If we consider making the least change or improvement, we have to go all over the basic plans of every manufacturer that we are selling and keep within that fence; sometimes there is less than 1/8-in clearance. The reverse-gear manufacturer has to make a product that can be sold commercially and to the largest number of engineers. When it gets down to making a special job for each one, it will make the expense too great. We are confined by the price and the cast-iron fence that the engine builder puts around the rear end. The reverse gear has to be manufactured to be commercially practical, but if we make any changes, those changes have to take into consideration every engine factory that we are selling. We have had discussions on changes that lasted for hours. We probably have from 75 to 100 different designs for reverse gears in our office. Some of them we consider ideal, but when we endeavor to provide each engine builder with a reverse gear that will meet his requirements, we are up against a pretty stiff proposition; so, we have to satisfy the majority of cases and not individual ones. If we make a change, that change must be such that it can go into every compartment of every engine builder we are selling.

CHAIRMAN HILL:—The motorboat industry has not appreciated that any clutch and any transmission will fit the rear end of any engine in an automobile or motor truck today because S. A. E. Standards are followed. The Society has endeavored many times to bring about such standardization in the motorboat field. It will save an immense amount of work, not only in the engine construction itself, but in the mounting of electrical starters, generators, clutches, reverse gears and other detachable units. The Society is ready to apply standardization to the motorboat field at any time the industry will join with us in establishing such standards and, moreover, will use them when once formulated.

MR. WESLEY:—Formerly, in our gear works, we built a different brake band and different after-bearing; the base would vary from 1 in. above the central line of the shaft to 1 in. below. We have been working on the elimination of different parts so that practically every brake band and after-bearing would be on the central line of the shaft; now, practically all of them were on that basis. if we can get the other standardization that we have been working on ourselves, and if the Society will influence the engine builders along the same line, much benefit for everybody will result.

MR. WARE:—The gear manufacturers are fairly close together now and very little work is needed to standardize them further so that a number of different gears could be dropped into the same case. We have looked into this very carefully in the last 2 years. The gears are fairly well standardized now for lengths and diameters; the main difficulty lies in the position of the cross-shaft bearing, that is the shaft which controls the clutches. When standardized in the manufacture of a gear-case cover that carries this cross-shaft bearing, it cannot be changed even 1/8-in. without spending a considerable sum of money on patterns and fixtures. It would seem to us that the reverse-gear manufacturers should get together with a view to keeping their new equipment interchangeable with one another.

Chairman Hill is right in saying that there should be more motorboat standardization. Our company stands ready to do anything it can along this line. It is highly desirable. We use the S. A. E. Standards in every possible way and many a design is thrown out because it is not adaptable to them.

COMMANDER S. M. KRAUS:—We have had cases where a single ship of the Navy was supplied with power boats that had engines of five different types and ignition apparatus of seven or eight different types. Where some of the engines were of the same manufacture but of different model with different ignition apparatus, it thus became necessary to carry spare parts for about three or four times as many engines as we should have. The result was that the Department has felt that it was almost impossible to buy its power-boat equipment in the open market. We do not want to build it, but we cannot keep it running if we have so many different kinds. I think that nothing would be more important to the industry than to standardize, although the standards should not be too rigid.

In reference to the relative cost of the light-weight and the medium-weight engines, sooner or later we will find that there is a field for the application of marine engines where the higher price of the light-weight engine will be

fully compensated for by the economy, not only due to the lower cost of operation, but also because much of that cost will be taken right out of the boat. When we really come to using marine engines in boats of long-distance cruising radius, I think we will find that we can very easily afford to pay the higher price that goes with the lighter-weight construction. I believe that the higher prices will be reduced materially in the future. Manufacturers will build either a smaller or a more accessible boat or one that has additional features that will sell the boat and its equipment to the owner who wants a pleasure boat or to the owner who wants a working boat, because those extra facilities will render the boat useful for a service that we otherwise do not attempt with the relatively small marine engine. We found that to be true in many directions in naval vessels.

Today, in cargo and passenger ships using internal-combustion engines, there is considerable reluctance to compare them with the cost of Diesel and other powerplants. Many have overlooked the chief difference and have compared the ships and their powerplants, ship for ship, but with different powerplants. I think the same situation will be found in the class of power boats and marine engines in which you are interested. Freight and passenger-steamship owners are discovering today that they must not compare a vessel of certain dimensions that is designed for one kind of power with the same size of vessel equipped with another type of power. You can build a radically different boat, get a very radically different service, apply it to radically different purposes and incur great additional expense, but with superior economy that you can build into the boat at the same time. I believe superior economy would cost nothing.

A. D. T. LIBBY:—I saw an engine at the Chicago automobile show that is, to my mind, the coming engine, not only in the marine field but in the farm-tractor field and in the automobile field itself. It answers many of the criticisms that have been directed toward the Diesel type of engine, one of which is that the Diesel engine is very heavy with respect to the amount of horsepower delivered. This new engine is of the semi-Diesel type and does away with many of the objectionable features of the Diesel engine, such as the air compressor and the air coolers, and the weight is very much less, comparatively speaking. I believe that it would be lighter than our present automobile engines for the horsepower delivered. The builder claims that this little engine will consume as little as 0.4 lb. of fuel per b.hp-hr., an enormous advantage; it is, therefore, an engine light in weight and high in fuel economy. None of the difficulties that Mr. Woolson mentioned with reference to carbureter and ignition troubles are encountered in this engine because it uses no carbureter and has no ignition system because it is of the high-compression type.

CHAIRMAN HILL:—It may straighten out a misapprehension held by some people to say that the Society's standardization work does not mean standardization to the point of telling a man what bore and stroke an engine should have, what type of water-jacket or any similar matter of design. The S. A. E. Standards relate principally to dimensions resulting in interchangeability.

J. G. VINCENT:—I believe our chairman, Mr. Hill, touched on a very important point when he compared the automobile industry with the motorboat industry in respect to interchangeability as a result of following standards. As very few of the marine engine builders manufacture their own reverse gears it would be a matter of great convenience and represent an important economic advantage if different makes of reverse gear could be applied to the same engine with no change in the reverse-gear housing and other important mounting dimensions. I would heartily indorse any such move toward standardization of this kind and would be glad to submit drawings of our own engines to representative reverse-gear manufacturers and any Society committee that may be formed to handle this subject.

DIRECT DETERMINATION OF DEW-POINTS OF GASOLINE-AIR MIXTURES

A METHOD of determining directly the dew-points of gasoline-air mixtures in the proportions required for use in internal-combustion engines has been devised by W. A. Gruse[1] and was presented to the division of petroleum chemistry of the American Chemical Society, in a paper read before that society at New Haven, Conn., on April 5, 1923. It is based on a belief in the fundamental significance of the dew-point of a gasoline-air mixture and consists of blowing a fuel mixture of known composition against an internally cooled metallic mirror and observing the temperature at which dew is formed.

A detailed description was given of the apparatus used in making the determinations and the results obtained were compared with those secured by R. E. Wilson and D. P. Barnard, 4th, by their method of equilibrium mixtures in tests previously reported by them.[2] The distillation curves of three commercial fuels bought in the open market at Pittsburgh were studied, their dew-points were investigated and the effects of adding 1 and 2 per cent of kerosene were observed with a view to determining the sensitiveness of the dew-point in the presence of small amounts of heavy ends. Increases of from 4 to 6 deg. in the temperature at which dew formed were noted with practically all of these fuels.

A direct determination of the dew-point of samples of the same fuels used and offered for test by Wilson and Barnard showed that their figures are approximately 20 deg. lower than those determined directly and that the change in the dew-point corresponding to a change from a 12 to 1 to a 15 to 1 mixture is of the order of 7 or 8 deg. when measured directly, whereas Wilson and Barnard in their tests previously referred to found a uniform variation of approximately 4 or 5 deg. for a number of different fuels.

A comparison of the figures for a second group of fuels shows that the direct determination gives dew-points that are higher in all cases than those arrived at by the equilibrium mixture. The direct method is said to embody the following features: (*a*) the apparatus can be constructed in an ordinary laboratory and machine shop; (*b*) with a little practice the dew-points can be read with fair accuracy and reproducibility; (*c*) it is believed to be sufficiently direct to be free from large errors; and (*d*) it applies to volatile fuels of any nature. It is offered tentatively as suitable for the direct determination of the "effective" volatility of motor-fuels with the idea that it may be useful in studying specifications and blending operations and for the control of other methods of evaluation.

The paper will be published in full in the August issue of the *Industrial and Engineering Chemistry*.

[1] Industrial fellow, Mellon Institute of Industrial Research, University of Pittsburgh, Pittsburgh.

[2] See THE JOURNAL, November, 1921, p. 313; January, 1922, p. 65; and March, 1923, p. 287.

Research Topics and Suggestions

THE Research Department plans to present under this heading each month a topic that is pertinent to the general field of automotive research, and is either of special interest to some group of the Society membership or related to some particularly urgent problem of the industry. Since the object of the department is to act as a clearing-house for research information, we shall be pleased to receive the comments of members regarding the topics so presented, and their suggestions as to what might be of interest in this connection.

MOTION AND FORCES BETWEEN WHEEL AND ROAD

THE subject of the so-called impact forces between the wheel and the road has been widely discussed of late and the results of experimental measurements have been published particularly by the Bureau of Public Roads. The subject, however, still seems to be rather an indefinite one in the minds of most engineers.

It is recognized in a general way that the tire characteristics, the speed and the ratio of the sprung to the unsprung weight are of prime importance, but just what relation they bear to the effect produced on the road by the vehicle seems not to have been very clearly analyzed. However, some phases of this problem are capable of definite mathematical expression and others can be stated approximately in mathematical terms.

One of the simplest cases of impact occurs when one or both wheels of a vehicle either drop from an obstruction or are projected into the air by going over an obstruction and then return to the road. General equations can be written which are applicable both to this and to other sorts of impact forces as well.

In considering this problem it is necessary first to clear up one point that is not always appreciated. This is the distinction between the *total energy* of an impact blow and the *maximum force* produced by it. When the blow is cushioned by a tire there is no fixed numerical relation between the two. The total energy is determined by the sprung and the unsprung weight and the spring constant for a given height of drop, while the maximum force depends upon the tire. It is the latter element alone that is of importance so far as the vehicle is concerned, and we believe of major importance also as regards the effect on the road.

The actions treated mathematically in a paper to be presented in the September issue of THE JOURNAL may be described briefly as follows: Consider the body and the axle at rest so far as vertical motions are concerned, with the springs under deflection carrying the load, as when traveling on a level road. The wheels are then allowed to drop freely for a specified distance as by running *rapidly* off an obstruction. The wheels are projected downward by gravity plus the force exerted by the springs. They therefore acquire a certain downward velocity and consequently an amount of kinetic energy equal to $\frac{1}{2}mv^2$ which is the total energy transmitted to the road by the wheels for this particular drop, and would be the same for pneumatic, solid or steel tires, except for a small difference readily accounted for.

The maximum force or load produced which represents the maximum load imposed on the road and on the axle itself, however, depends entirely upon the distance in which and rate at which this amount of energy is transmitted to the road; in other words, it depends upon the total depression of the tire from the instant it strikes the road to the instant of greatest depression. This total force is the sum of (*a*) the impulsive force represented by the maximum rate of deceleration of the wheels which coincides practically at least with the maximum tire depression, (*b*) the weight of the axle, and (*c*) the static force applied by the springs at this instant. Therefore, if it is assumed that the depression of the tires is the same under impact as when measured statically the maximum force can be calculated from the known static depression curve of the tire. The extent to which this assumption is justified has not been determined definitely but we believe it is qualitatively correct and that the conclusions which can be drawn from the results of the forthcoming paper may be of value as a guide to the relative effect of the tire deflection, the height of the drop and the relation of the sprung and the unsprung weight on the maximum forces to which wheels and axles as well as roads are subjected by this type of impact.

To illustrate the application of the results of the analysis as it will appear next month, if it is assumed, for example, that the height of fall from an obstruction is 3 in. the time of fall is found to be approximately 0.060 sec. and 0.075 sec. for ratios of sprung to unsprung weights of 4 and 2 respectively; in these cases a constant spring deflection of 4 in. under the weight of the sprung mass and a constant total weight are assumed. It is also found that the velocity of impact of the unsprung mass is 6.84 ft. per sec. for the first case and 5.35 ft. per sec. for the second; the kinetic energy is 4690 ft.-poundals for the first ratio of 4, and 4770 ft.-poundals for the second ratio of 2, while the impact forces are 2480 and 2590 lb. for the respective cases. Again keeping all conditions the same, except the tires, and using a ratio of sprung to unsprung weight of 4, the forces are 2050 lb. for the low-pressure pneumatic tires and 2480 lb. for the high-pressure pneumatic tires.

Thus the change of impact force due to the change in tires is about 25 per cent while that occasioned by doubling the ratio of the sprung to the unsprung weight is only about 4 per cent. Changes in the type of tire have more influence on the impact force than changes in the weight ratio.

OBITUARY

JAMES FREDERIC BOURQUIN, vice-president and general manager of the Continental Motors Corporation, Detroit, and for years an outstanding figure in Detroit's automobile industry, died, July 1, 1923, in a hospital in that city, following an operation for appendicitis, at the age of 45 years. He was born, April 9, 1878, at Detroit and, following his public and high school education, entered the University of Michigan from which he was graduated in 1904 with the degree of Bachelor of Science.

Mr. Bourquin's practical experience began in the experimental department of the Old's Motor Works, Lansing, Mich., in 1904, and was continued progressively until 1910 with this and other companies in capacities as draftsman, foreman, inspector, assistant superintendent and superintendent of manufacturing, the last being with the Chalmers Motor Co. He was active in organizing the Liberty Motor Car Co. and also contributed largely to the development of the Paige-Detroit Motor Car Co., both of Detroit; in 1911 he was general manager of the latter company.

Among the clubs to which he devoted time and enthusiasm were the Detroit Athletic Club and the Detroit Yacht Club. Also, he was a member of several clubs in Ann Arbor, Mich., and was an ardent Knight Templar. He was elected to Member grade in the Society, Jan. 30, 1911.

SPARK-ADVANCE IN COMBUSTION ENGINES

(*Concluded from p. 121*)

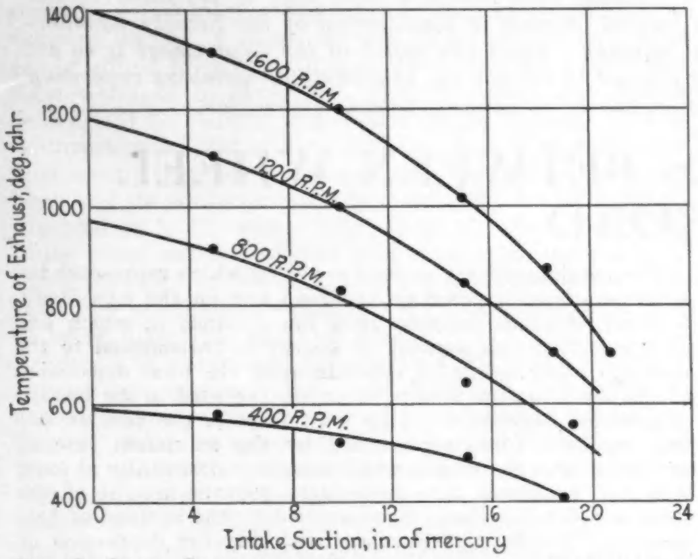


FIG. 17—CURVES SHOWING THE RELATION BETWEEN THE TEMPERATURE OF THE EXHAUST AND THE INTAKE SUCTION OR LOAD IN A CONTINENTAL SIX-CYLINDER ENGINE

The fairing of the data is done in Fig. 17; Fig. 18 is only a replotting of the faired lines of Fig. 17, with the variables exchanged.

We repeated on the Continental engine the findings on the Ford; that for all practical purposes the optimum spark-advance can be represented as the sum of two functions, one a function of speed only, the other a function of intake suction only. This follows from the fact that the curves in Fig. 15 are substantially parallel; the increase of the spark-advance for a change of the intake suction at a constant speed is shown in the lower curve that is marked intake-suction effect, in Fig. 15. An approximate empirical equation for this intake-suction effect is $a = 18\ (p_e/P_i)$. For the Ford engine the corresponding equation is $a = 13\ (p_e/P_i)$; the effect is less, probably, because the dilution at a zero intake-suction is larger in the lower-compression engine.

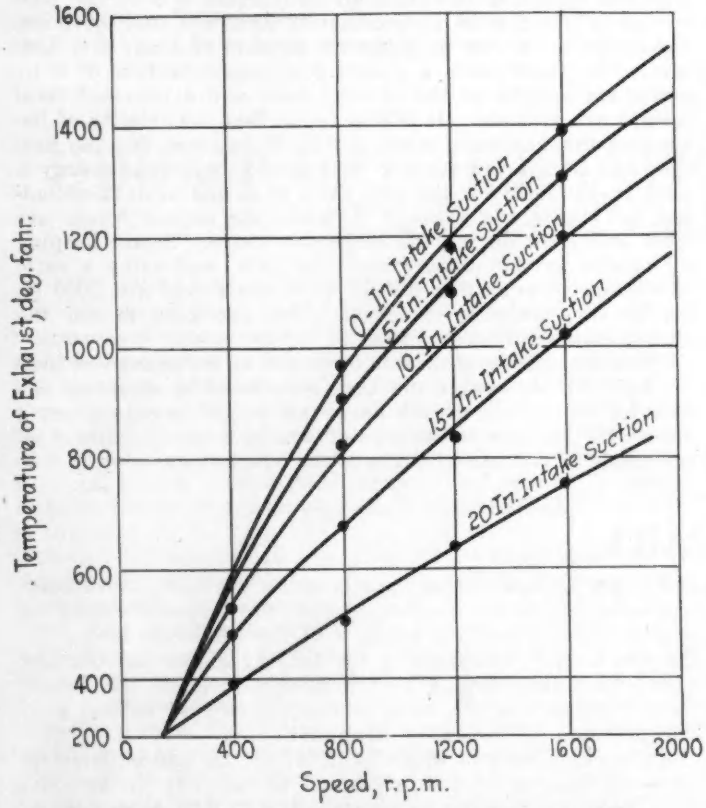


FIG. 18—CURVES SHOWING THE RELATION BETWEEN THE TEMPERATURE OF THE EXHAUST AND THE SPEED OF A CONTINENTAL SIX-CYLINDER ENGINE FOR DIFFERENT CONSTANT INTAKE-SUCTIONS OR LOADS

The power losses from improper spark-advance and the economy losses in proportion are clearly indicated in Figs. 11 to 14. It seems, in view of these curves, that automatic spark-control by mechanically adding the two separate advances for speed and for intake suction would justify itself in improved power, economy and flexibility of engine performance. There would still be need of some hand adjustment to take care of a cold engine and of a dirty engine; but with the engine warmed up and a setting made for its internal condition all the spark-control needed to take care of the load and the speed variation would be automatic and would be done as now it is not. It would also be easy to take care of altitude effects by changing the linkage on the automatic spark-control on the intake-suction side; this would be necessary because of the greater dilution with exhaust gas at high altitudes for the same intake-suctions. The automatic device would be controlled by pressure differences, whereas the function it controls is really a consequence of pressure ratios; $p_e - P_i$ could be substituted in the mechanism for p_e/P_i.

In a scientific analysis of that component of spark-advance due to dilution with exhaust gas we need to find whether it is a matter of volumetric or of mass dilution, and whether in computing the dilution we must figure the re-expansion of the exhaust gases in the suction stroke as adiabatic or isothermal. It was to get data for this study that the readings of the exhaust temperatures in Figs. 11 to 14 and in Figs. 17 and 18 were taken. In previously discussing the dilution matter in this paper the formulas for the dilution-ratios have been derived and the probable result has been predicted that the spark-advance due to dilution will be found to be an exponential function of the mass-dilution ratio.

The volumetric-dilution ratio of the total gas to the new charge in the cylinder is $1 + (p_e/P_i)^{1/y}/[r - (p_e/P_i)^{1/y}]$; the weight-dilution ratio is $1 + ([T_i/T_e]\ [p_e/P_i]/[r - (p_e/P_i)^{1/y}]$. In these formulas r is the compression-ratio, 4.55 for the Continental engine; y is the ratio of the specific-heats, 1.33 for adiabatic and 1.00 for isothermal expansion; $1/y = 0.75$ or 1.00; T_i and T_e are the *absolute* temperatures in the intake and the exhaust manifolds; T_i is taken as $100 + 460 = 560$ deg. fahr.; absolute, because of the "hot-spotting" of the manifold; p_e and P_i are the absolute pressures in the exhaust and intake manifolds. In calculations p_e was taken as 29.4 in. of mercury and P_i as $(p_e - S)$, S being the intake suction in inches of mercury. Toward full throttle, the variation of the dilution with p_e/P_i is slow; at closed throttle the exhaust back-pressure is negligible; hence the use of the barometric pressure divided by the difference between the barometer and the intake-suction readings for p_e/P_i causes little error in the calculations. The calculated volumetric and mass-dilution ratios for the Continental engine are tabulated in Table 4.

With the values of the volumetric and the mass-dilu-

TABLE 4—CALCULATED VOLUMETRIC AND MASS-DILUTION RATIOS FOR A SIX-CYLINDER CONTINENTAL ENGINE

Intake Suction	Intake Pressure		Volumetric Dilution Ratios		Mass Dilution-Ratios, Adiabatic, Speeds, R. P. M.				Mass Dilution-Ratios, Isothermal, Speeds, R. P. M.			
S	P_1	p_e/P_1	Adiabatic	Isothermal	400	800	1,200	1,600	400	800	1,200	1,600
0	29.4	1.000	1.28	1.28	1.15	1.11	1.10	1.09	1.15	1.11	1.10	1.09
5	24.4	1.206	1.34	1.36	1.19	1.15	1.13	1.11	1.19	1.15	1.13	1.12
10	19.4	1.516	1.43	1.50	1.27	1.21	1.18	1.16	1.28	1.22	1.19	1.17
15	14.4	2.042	1.60	1.81	1.43	1.36	1.31	1.27	1.49	1.40	1.35	1.31
20	9.4	3.127	2.07	3.20	1.93	1.83	1.72	1.66	2.44	2.27	2.12	2.02
22	7.4	3.975	2.62	7.92	...	...	...	...	...	...	...	...

tion ratios in Table 4, and the optimum spark-advances from the faired curves in Figs. 15 and 16, curves of spark-advance against dilution-ratios for constant speeds have been plotted in Fig. 19 on logarithmic cross-section paper. If dilution had an effect that is separable from that of speed or turbulence, the family of curves for the different constant speeds in these charts should be a set of parallel lines of constant vertical offset from each other. This is obviously not the case when we measure dilution volumetrically; it is approached closely when we measure dilution by mass ratio and is given better by the adiabatic than by the isothermal assumption of the law of re-expansion of clearance, or exhaust, gases on the suction stroke. Further, the dashed straight lines of the two charts at the right show that the increase of spark-advance due to dilution is approximately proportional to the cube of the mass-dilution ratio; or, in other words, the combustion rates vary inversely and the explosion-times directly as the cube of the adiabatic mass-dilution ratio of the total gas to the new charge. If in plotting the curves at the extreme right the unfaired data of the optimum spark-advances, taken directly from Figs. 11 to 14, is used, the agreement with the cube law is rather improved; hence the use of our faired curves of spark-advance has not been responsible for this suggested law of the action of dilution. This cubic law, of course, is merely suggested, not proved, by such a slender basis of data as is used here; but it was to be expected on general grounds and has been found here to account for the facts, as well as any simple hypothesis that has been tried.

Evaluation of the Turbulence Factor

Since it is now possible to eliminate the dilution or intake-suction factor from the spark-advance, the turbulence factor may be evaluated. This has been done in two ways. First, as was done in the case of the Ford, all the spark-advance data were reduced to values for a zero intake-suction at the various speeds. Fig. 16 shows this reduction for the Continental engine. Then values of R/a_o, or a zero intake-suction, were plotted against R, giving the lower straight line in Fig. 20. The equation of this line is $R/(a_o) = 22\ (1 + 0.001R)$; or by inversion, a_o is equal to $0.0455R/(1 + 0.001R)$. This corresponds to an explosion-time in the combustion-chamber without turbulence, but with the dilution exist-

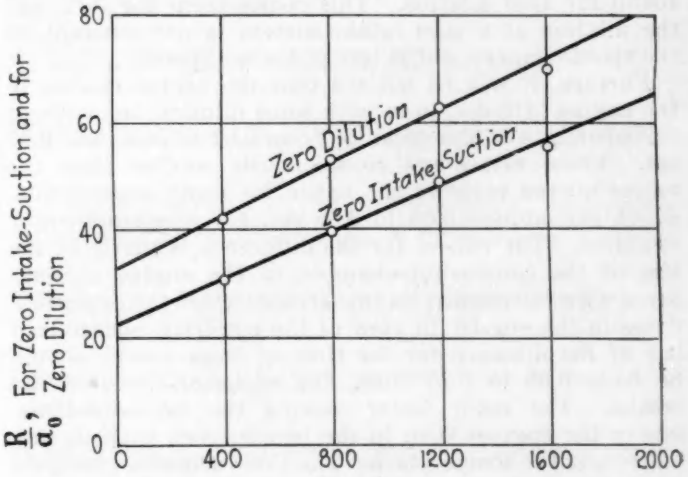


FIG. 20—CURVES OF ZERO DILUTION AND ZERO INTAKE-SUCTION AT VARIOUS SPEEDS

ing at a zero intake-suction, of $(2/9) \times 0.0455 = 0.0101$ sec. This may be compared directly with the corresponding value for the Ford engine of 0.0230 sec.; and the turbulence factor, found in the same way, comes out the same, 0.0010, for both engines. This is interesting as both are of the L-head type and of rather similar dimensions. The much quicker explosion-time for the Continental engine may correspond to a better shape of the combustion-chamber, a better placing of the spark-plug, better carburetion due to the hot-spot and the like and the higher compression-ratio.

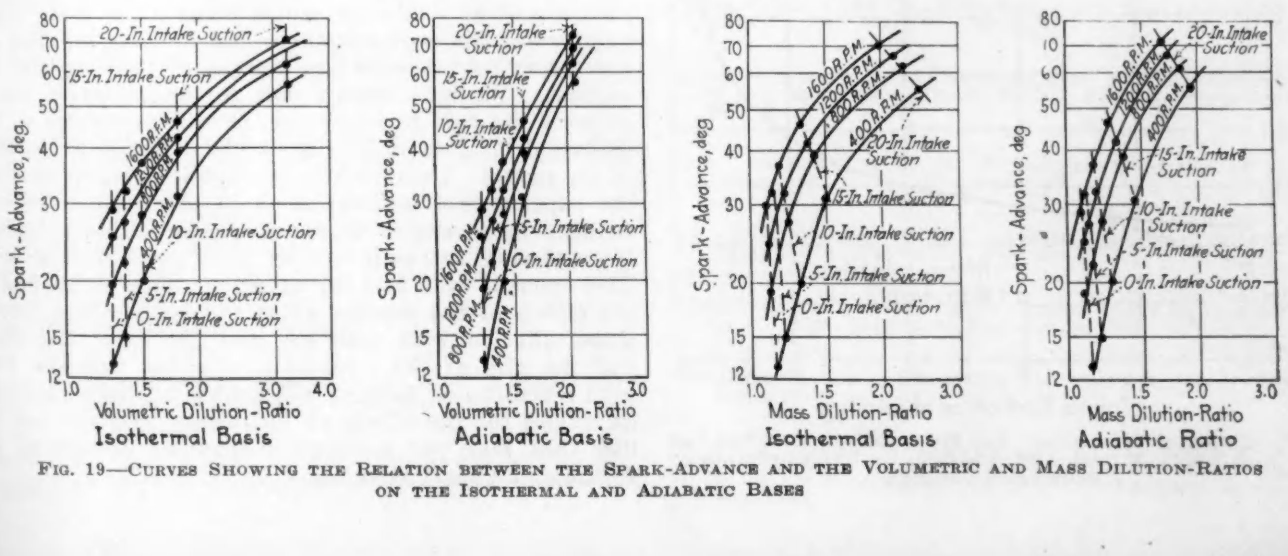


FIG. 19—CURVES SHOWING THE RELATION BETWEEN THE SPARK-ADVANCE AND THE VOLUMETRIC AND MASS DILUTION-RATIOS ON THE ISOTHERMAL AND ADIABATIC BASES

It is possible, also, in the case of the Continental engine, to investigate the case of zero dilution. Assuming the cubic law of the slowing-up of the combustion by adiabatic mass-dilution, the spark-advances for the Continental engine at zero dilution are 9.5 deg. at 400 r.p.m., 15 deg. at 800 r.p.m., 19 deg. at 1200 r.p.m., and 23 deg. at 1600 r.p.m. (See the right half of Fig. 19 for this reduction to zero dilution, or a 1.00 dilution-ratio.) The upper straight line in Fig. 20 plots R/a_0 of these zero-dilution values against R. Its equation is R/a_0, or zero dilution, $= 32.5\ (1 + 0.00075R)$ and corresponds to a_0 for zero dilution $= 0.0308R/(1 + 0.00075R)$, and the explosion-time with zero dilution and zero turbulence of $(2/9) \times 0.0308 = 0.0068$ sec.

It will be noticed that the value of the turbulence factor, the coefficient of R in the denominator of the spark-advance equations, is not the same when the equation is found for a zero intake-suction as it is when found for zero dilution. This comes from the fact that the dilution at a zero intake-suction is not constant as the speed changes, but is less at higher speeds.

Further, it will be noticed that the explosion-time in the engine cylinder, even with some dilution but without turbulence, is of the order of from 0.02 to less than 0.01 sec. These values are conspicuously smaller than the values quoted early in the paper for bomb experiments, which ran around 0.05 to 0.06 sec. for maximum-power mixtures. The reason for the difference is partly in the size of the combustion-chamber of the engine as compared with the bombs; on this account alone the explosion-times in the engine, in view of the empirical square-root law of the distance for the time of flame travel, should be from 0.50 to 0.75 times the explosion-times in the bombs. The main factor making the explosion-times less in the engines than in the bombs, even without turbulence, is the temperature. The Continental engine gets its charge at about 560 deg. fahr., absolute. Compression multiplies this temperature by the $(y\text{-}1)$ power of the compression-ratio, or by about 1.65, making the temperature preceding ignition about 930 deg. fahr., absolute. As the combustion speed is approximately proportional to the cube of the temperature preceding ignition, the combustion-rate is increased by adiabatic compression about 4.5 times over what it would be at the intake temperature and about 5.3 times over what it would be at the room temperature. Even without turbulence, then, we might expect the explosion-times in automotive engines to be of the order of 1/5 to 1/10 of what they are in the bomb experiments cited. The effect of turbulence is to offset the slowing-up of the combustion due to dilution, rather than to cause the explosion-times in the engine cylinder to be less than in the experiments on sizable bombs in laboratory work. Of course, turbulence remains highly important in high-speed engines.

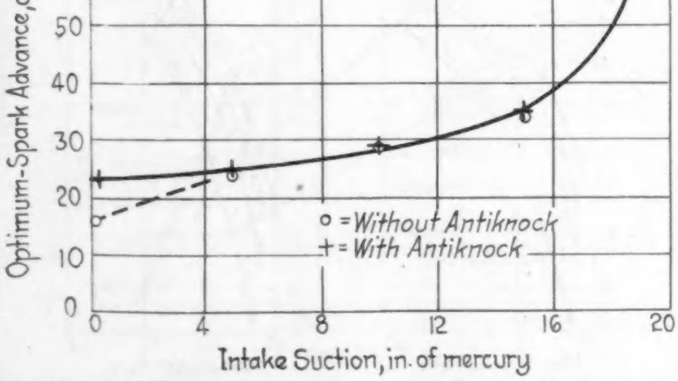


Fig. 21—Effect of Detonation and Its Suppression by the Use of an Anti-Knock Material on the Optimum Spark Advance of a Continental Engine

In the experimental work on the optimum spark-advance on the Continental engine, shown in Figs. 11 to 14, we did not get the data at full throttle but went only as close as the 5-in. suction. This was because of the detonation that occurred at pressures lower than the 5-in. intake-suction. We investigated later what would happen in the presence or absence of detonation. The work was at 600 r.p.m. The results are shown in Fig. 21. We ran through the whole throttle range, 0, 5, 10, 15 and 19-in. intake-suctions, without anti-knock and then with it. The dose of anti-knock, tetraethyl-lead, was 5 cc. to 2 qt. of gasoline, or 20 times the dosage usually required or advised for knock suppression.

Discussion of Results

The results are extremely interesting. For any intake-suction large enough of itself to stop detonation by throttling or pressure-reduction, the presence of anti-knock, in a dosage 20 times the usual value, does not change the optimum spark-advance at all, according to our data; hence it does not affect the explosion-time or the reaction-velocity of combustion. The accuracy of determining the optimum spark-advance is perhaps $\pm$ 2 deg.; on a 30-deg. advance this is an accuracy in the explosion-time of $\pm$ 7 per cent. The usual dosage of anti-knock, then, cannot alter the reaction-rate by more than 0.33 per cent. The hypothesis that anti-knocks stop detonation by changing the reaction-rate of combustion seems to have here a considerable jolt. It does stop detonation, however; and further, with detonation stopped, the spark-advance-intake-suction curve is smooth and continuous to the zero intake-suction; and combustion follows a single set of laws, those of normal non-detonating combustion, over the whole throttle-range. In the absence of anti-knock, combustion seems to follow two laws, as indicated by the dotted line in Fig 21; the normal laws for intake-suctions greater than 5 in. in this case; and the abnormal laws, involving detonation, at between zero and a 5-in. suction. When detonation occurred the torque also was perceptibly less than when it was suppressed.

In summarizing, we have shown that spark-advance is a relation of the explosion-time to the speed of rotation of the engine. From a simple hypothesis about flame speed, combined with the geometric shape of the combustion-chamber, we have deduced that one-half the rise of pressure of an explosion should occur at or near three-quarters of the explosion-time; and we have checked the conclusion on pressure-time curves of both bombs and engines. We have shown that for an optimum spark-advance the half pressure-rise, or three-quarters of the explosion-time, should occur at the dead-center position of the piston. Hence, we have made it possible, through the experimental determination of the optimum spark-advance in an engine in operation, to calculate back to the explosion-time as it actually takes place in the engine. This opens a wide field for experimental work in finding the effects of the mixture-ratio, the temperature, turbulence, dilution with dead gas and the like,; the shape and the size of the combustion-chamber and the like; with a possibility, hitherto not known, of finding numerical values for the effects of turbulence, dilution and the like that may put combustion-chamber design of the future on a quantitative basis.

Applicants *Qualified*

The following applicants have qualified for admission to the Society between June 10 and July 10, 1923. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

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Applicants *for* Membership

The applications for membership received between June 15 and July 13, 1923, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ARCHER, T. P., factory manager, Ternstedt Mfg. Co., *Detroit.*

BLACKEBY, F. WALTER, manager of sales, electric steel department, Eastern Steel Castings, Inc., *Newark, N. J.*

COBB, FRANCIS S., president, Seamans & Cobb, *Boston.*

DUST, JOHN G., experimental foreman, Phelps Light & Power Co., *Rock Island, Ill.*

EDGAR, F. M., chief engineer, Ternstedt Mfg. Co., *Detroit.*

GILPIN, BENJAMIN H., factory manager, Standard Steel and Bearings, Inc., *Plainville, Conn.*

GUILLELMON, M., vice-president, Renault Selling Branch, Inc., *New York City.*

GUSTAFSON, A. A., works manager, Yellow Sleeve Valve Engine Works, Inc., *East Moline, Ill.*

HOWELL, MILLARD T., electric vehicle engineer, Bureau of Power & Light, *Los Angeles.*

HYATT, WILLIAM D., layout draftsman, Durant Motors of Indiana, *Muncie, Ind.*

IRVINE, DAVID M., chief inspector, Durant Motors Corporation, *Elizabeth, N. J.*

JORGENSON, CLARENCE H., consulting engineer, Dole Valve Co., Chicago, and mechanical engineer, Jorgenson Engineering Co., *Chicago.*

KIDDER, EDWIN H., general sales manager, Dunlop Tire & Rubber Co. of America, *Buffalo.*

LAWRENCE, WILLIAM A., president and master mechanic, W. A. Lawrence & Co., *New York City.*

LOUGHEAD, MALCOLM, chief engineer, Four Wheel Hydraulic Brake Corporation, *Detroit.*

LUKER, FRED, mechanical engineer, General Motors Research Corporation, *Dayton, Ohio.*

MCCADDEN, M. H., secretary, Continental Piston Ring Co., *Memphis, Tenn.*

MEESE, HORACE S., transportation consultant, Commercial Truck Co., *Philadelphia.*

NEALE, FRED, designer, Northway Motor & Mfg. Co., *Detroit.*

OESTERMEYER, CARL F., electrical engineer, Willard Storage Battery Co., *Cleveland.*

PRATT, THEODORE D., civil engineer and general manager, Motor Truck Association of America, *New York City.*

RAUEN, CARL F., aeronautical draftsman, McCook Field, *Dayton, Ohio.*

RUSHMORE, S. W., owner, Rushmore Laboratory, *Plainfield, N. J.*

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USHIODA, SEIKICHI, 2-2 Mita, Shiba-ku, *Tokyo, Japan.*

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WINZENBURG, FRED C., commercial artist, Winzenburg Artists, *Chicago.*